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SVHSER 6180

ANALYSIS AND TEST OF A BREADBOARD CRYOGENIC
HYDROGEN/FREON HEAT EXCHANGER

by

L. F. DESJARDINS and J. HOOPER

PREPARED UNDER CONTRACT NUMBER NAS 9-12725

by

HAMILTON STANDARD
DIVISION OF UNITED AIRCRAFT CORPORATION
WINDSOR LOCKS, CONNECTICUT

for

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

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ABSTRACT

ANALYSIS AND TEST OF A BREADBOARD CRYOGENIC
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L. F. Desjardins and J. Hooper

April 1973

This report describes the system studies required to verify a tube - in - tube cryogenic heat exchanger as optimum for the Space Shuttle mission. Design of the optimum configuration, which could be fabricated from commercially available hardware, is discussed. Finally, testing of the proposed configuration with supercritical hydrogen and Freon 21 is discussed and results are compared with thermal and dynamic analysis.

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FOREWORD

This report was prepared by the Hamilton Standard Division of the United Aircraft Corporation for the National Aeronautics and Space Administration's Manned Spacecraft Center in accordance with Contract NAS 9-12725. This report covers the period from May 12, 1972 through January 24, 1973. During this period the tube - in - tube was selected as the optimum cryogenic heat exchanger configuration for the Space Shuttle. A breadboard unit was designed, fabricated and tested to correlate analysis with actual performance.

Personnel responsible for the conduct of this program were Mr. F. H. Greenwood, Program Manager, Mr. L. F. Desjardins, Engineering Project Manager, and Messrs. J. Hooper and R. Balinskas, Analytical Engineers. Testing was conducted at Wyle Laboratories at Norco, California. Mr. F. Collier was Technical Monitor and Mr. R. J. Gillen overall program supervisor for the NASA Manned Spacecraft Center.

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SUMMARY

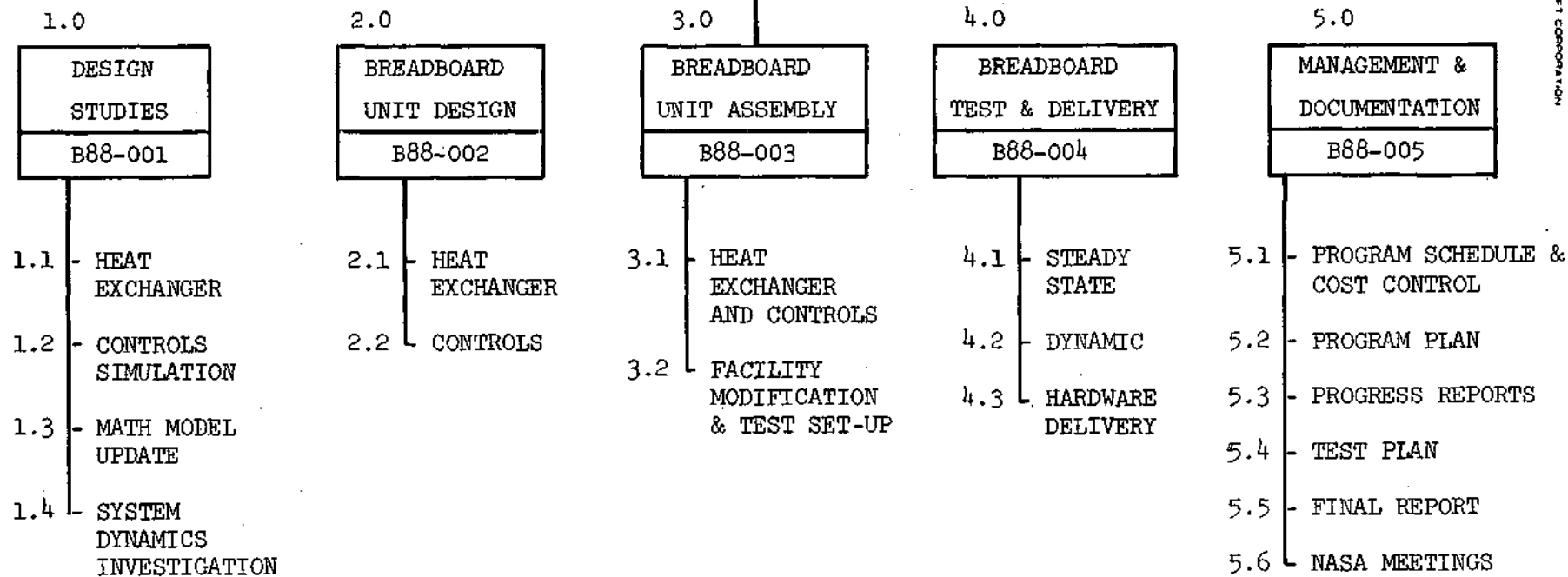
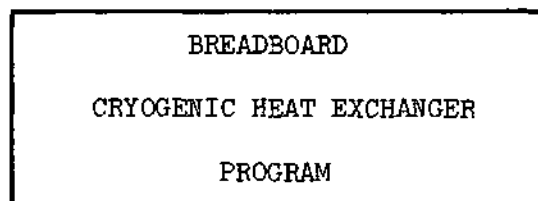
Initial studies indicate that the Space Shuttle system heat rejection requirements may be optimized by the use of a cryogenic heat exchanger. The eventual viability of this heat rejection method is dependent upon the technology development of a heat exchanger concept for use with cryogenic hydrogen and Freon 21. A simple tube - in - tube concept, selected for the Space Shuttle Orbiter under contract NAS 9-12208, showed promise of providing the required features at a very low weight penalty and providing basic conceptual simplicity. This concept had been modeled mathematically and its feasibility demonstrated by tests conducted with cryogenic nitrogen and water.

The objective of this phase II program was the correlation of analytically derived data with test data from a heat exchanger utilizing the Shuttle fluids of supercritical hydrogen and Freon 21. As can be seen in the Work Breakdown Structure (WBS), figure 1, the program was divided into 5 major tasks, which were sequential except for the generalized task of Management and Documentation which occurred throughout the program. At the outset, the latest available Shuttle data was examined and found similar to that on which the previous phase was based. Thus the same rationale supporting a tube - in - tube cryogenic heat exchanger was still applicable, as well as the proposed control system. Next, the heat exchanger was optimally sized using commercially available tubing. Aluminum was selected because of its superior physical properties⁽¹⁾ at cryogenic temperatures, high conductivity and minimum weight. The design selected, namely concentric 1" O.D. and 3/4" O.D. concentric tubes, 15 feet long in a coil, approximately 1 foot diameter, was included in the math model and computer program previously prepared. System dynamic studies showed the proposed cryogenic heat exchanger control system both accurate and stable.

The proposed heat exchanger was defined in drawing SVSK 86021 and a vendor was selected for its manufacture after receiving NASA design approval. Controls of the proposed cryogenic heat exchanger control loop were not definitized in drawings since their procurement or test were not a part of the program. Considerable difficulty was encountered in manufacture of the heat exchanger in that the maximum length of commercially available tubing was 12 feet, requiring a butt welded joint which fractured during coiling. In addition severe rippling of the outer tube was experienced. To correct these problems the wall thickness of the outer tube was increased from .028" to .035" and the tubing length was limited to 12 feet. Future prototype procurement should provide time and funding for a mill run of the proper length tubing.

Wyle Laboratories in Norco, California was selected to conduct the heat exchanger tests after soliciting bids from other test facilities and examining Hamilton Standard's test capabilities. Wyle was particularly

⁽¹⁾ALCOA Bulletin "ALCOA Aluminum The Cryogenic Metal."



WORK BREAKDOWN STRUCTURE

FIGURE 1

advantageous in that they had a complete functional cryogenic hydrogen facility and took no deviations to the test plan. Steady state thermal performance tests were conducted at 5000, 20,000, 35,000, and 50,000 Btu/hr. Transient tests were conducted by increasing Freon inlet temperature from minimum to maximum in step and ramp changes and starting and stopping hydrogen flow at 5000 and 50,000 Btu/hr heat loads. Good correlation with analytically derived thermal and dynamic performance was obtained ; however, a progressively more severe hydrogen instability was noted with increasing heat load. This was manifest as a one hertz fluctuation in hydrogen inlet and outlet pressure and flow. Amplitude of the pressure fluctuations at the 50,000 Btu/hr case was $\pm 15-20$ psi. Investigation confirmed that the hydrogen pressure/flow fluctuations were not caused by the test rig.

Literature searches revealed a considerable number of similar instabilities but the mechanism of instability has been postulated only. Some show it as a function of low cryogenic flow rates and high heat loads and others as a function of cryogenic inlet to outlet density ratios. Clearly, however, the mechanism of instability must be understood better before using a system with the inherent instability or attempting to redesign a heat exchanger for stability.

INTRODUCTION

This program was conducted in accordance with NASA Contract NAS 9-12725 to verify the selection of a tube - in - tube configuration as the optimum cryogenic heat exchanger for the Space Shuttle program, and design, manufacture and test a breadboard unit. This program is a follow-on to NASA Contract NAS 9-12208 which authorized conduct of an in depth study to develop a cryogenic heat exchanger concept meeting Space Shuttle requirements of simplicity, light-weight, reusability, low maintenance, long life and low cost. This program recommended a tube - in - tube counterflow configuration on the basis of lower cost and longer life than the other primary contenders of tube - in - shell and plate and fin. System weight difference among these three primary contenders was only 3.6 of 121.4 pounds hence the small weight advantage of the plate and fin configuration could not counteract the cost and life advantages of the tube - in - tube configuration.

The major tasks of this program consist of the following:

Design Studies - These studies reviewed established results with latest Shuttle conditions and provide further math model development and system dynamic evaluation.

Breadboard Unit Design - This task established a tube - in - tube configuration which provided minimum system weight consistent with commercially available tubing and fittings.

Breadboard Unit Fabrication - A prototype heat exchanger was fabricated.

Breadboard Tests - Steady state and transient tests were conducted within a range of 5000 to 50,000 Btu/hr to afford correlation with the analytical approach for dynamics and thermal performance.

CONCLUSIONS

- The tube - in - tube heat exchanger is confirmed the optimum configuration for Shuttle cryogenic application.
- Math model and computer studies have shown that the previously selected thermal control system provides the required system stability and accuracy.
- Test data verifies thermal and dynamic analysis of the heat exchanger math model.
- The cryogenic heat exchanger as tested displays a hydrogen pressure and flow oscillation of approximately one hertz which increases in magnitude with increasing heat loads.
- The thermal control system proposed for use with the cryogenic heat exchanger meets thermal stability and accuracy requirements despite hydrogen oscillations of the selected heat exchanger.
- The mechanism of hydrogen oscillations is neither completely understood nor predictable.

RECOMMENDATIONS

- Further studies and tests should be conducted to:
 - Understand the mechanism of cryogenic hydrogen oscillations to a degree permitting stable heat exchanger design.
 - Understand the characteristics and implications of such oscillations in a Shuttle cryogenic fluid loop.
- Mill runs should be procured in the future to permit a heat exchanger length resulting in optimum system weight.
- For the tubing sizes selected in this study coil size should be a minimum of 12 inches I.D. to prevent rippling.

DISCUSSION

The discussion is divided into four major sections following the program work breakdown structure. These sections are Design Studies, Unit Design, Unit Assembly, and Unit Test and Delivery.

DESIGN STUDIES

A review of the cryogenic heat exchanger requirements and system penalties was completed. Contact with North American Rockwell showed that heat loads and cryogenic system penalties had not changed. Thus the tube - in - tube configuration determined best under NASA/MSC contract NAS 9-12208 continues as the selected configuration. An optimization of the tube - in - tube heat exchanger size and weight using commercially available aluminum tubing was conducted. This optimization considered a maximum heat load of 50,000 Btu/hr, and normal heat loads, zero load operation, hydrogen utilization, hydrogen supply pressure variations, degree of hydrogen super-heat and coolant freezing prevention. In addition, nominal loads were assumed for gross structural investigation as specified in figure 3.

The cryogenic heat exchanger was optimized based on the following condition:

T of Freon 21 in	=	620°R
T of Freon 21 out	=	500°R
T of H ₂ in	=	66°R
P of H ₂ in	=	300 psi
Q	=	50,000 Btu/hr
W of Freon 21	=	1602.5 lbs/hr
Usage Time	=	1 hour

This condition is based on North American Rockwell data. Power penalty to account for the Freon 21 pressure drop was calculated as 2.39 lbs/psi based on the North American Rockwell system data.

The pipe outside diameters that were used in the optimization were 3/8" to 1" in commercially available 1/8" increments between these limits. Thicknesses investigated were .020, .028, and .035 inches. As aluminum is

compatible with both hydrogen and Freon 21 and has excellent mechanical properties at cryogenic temperatures, it was selected as the heat exchanger material. Due to the long tube length, conduction along the walls is insignificant.

Figure 2 compares the following combinations of pipe and wall thickness:

1. 1/2", .028; 3/4", .028"
2. 3/4", .028; 1", .028"
3. 5/8", .028; 7/8", .028"
4. 3/8", .020; 5/8", .020"
5. 3/4", .035; 1", .035.

Total equivalent weight consists of:

- a) Heat exchanger "wet weight".
- b) Hydrogen expended for one hour.
- c) Freon pumping power penalty.
- d) Tankage penalty was not included.

The difference in total equivalent weight for the configurations compared was less than two pounds at their optimum lengths. Configuration 2 was chosen with a length of 15 feet as it has the lowest total equivalent weight.

All of the configurations examined had the minimum wall temperatures above 410°R with the exception of configuration 4. As configuration 2 was selected there is no danger of freezing the Freon 21. Freon 21 freezes at 249°R.

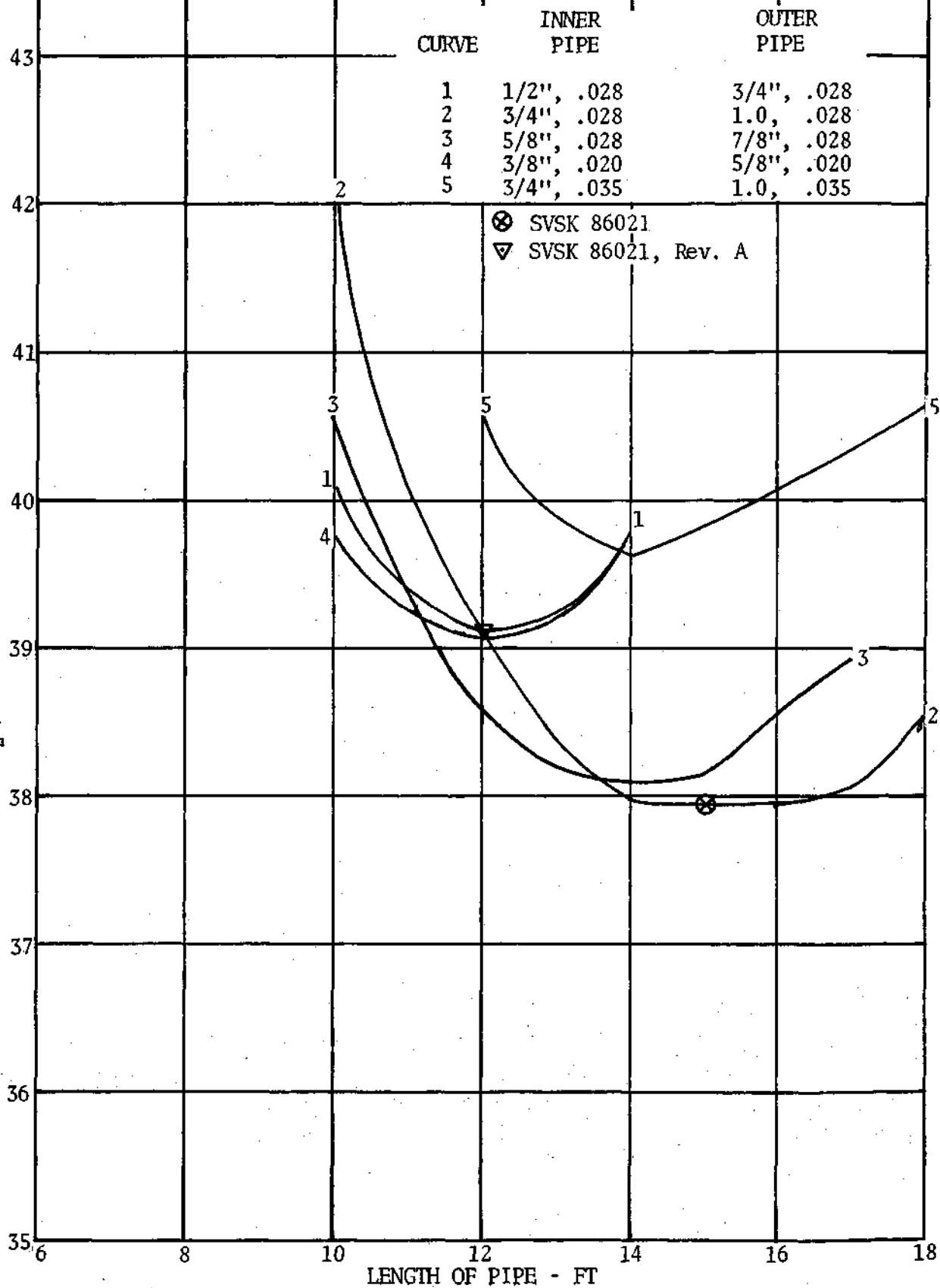
Variations in H₂ supply pressures do not affect the heat exchanger size and were not considered in heat exchanger sizing.

For comparison, a plate fin heat exchanger was sized to determine the weight saving that could be achieved. A maximum weight saving of 3.6 pounds was calculated using a plate fin heat exchanger. It was concluded that this weight saving did not justify the increased cost and development required for a plate fin heat exchanger.

The stresses of the coiled design were investigated for:

- a) Thermal stresses.
- b) Vibrational stresses.
- c) Shock.

TOTAL EQUIVALENT WEIGHT - LBS
 W_{H_2} Lbs/hr + W_{wet} + ΔP FREON x 2.39)



CRYOGENIC DOUBLE PIPE HEAT EXCHANGERS
FIGURE 2

The configuration, mounting positions, and requirements are shown in figure 3. The conclusion was reached that there are no structural problems in the design.

Controls Simulation and Math Model Update

The analytical model for the dynamics of a cryogenic tube - in - tube heat exchanger, which had previously been computerized, was updated to include the results of the optimization studies and was expanded to include the dynamics of the Shuttle Freon loop.

The Freon loop parameters for this investigation were selected on the basis of recent information obtained from North American Rockwell Space Division as well as in-house studies. Figure 4 shows the design heat loads selected for the active components and loop temperatures. Figure 5 shows line lengths and the weight of all components in the loop. The dynamic parameters for the model are summarized in Table I.

The computerized model includes all elements in the radiator bypass loop, but not the radiator, hydraulic heat exchanger and associated lines. The cabin water loop line from the interface heat exchanger to the cabin condensing heat exchanger has also been included in order to obtain the effect of Freon loop temperature fluctuations on the temperature of cooling water entering the condensor.

System Dynamics Investigation

A mathematical model of a tube - in - tube cryogenic heat exchanger was prepared under Contract NAS 9-12208. A controls investigation made using the computerized model was reported in NASA CR 115569. In that report a recommendation was made for a simple control, sensing Freon outlet temperature, and employing a constant speed reversible electric motor to actuate the H₂ feed valve. In the absence of detailed knowledge of the thermal dynamics of the Shuttle Freon loop, characteristics for the control were selected on the basis of assumed values for:

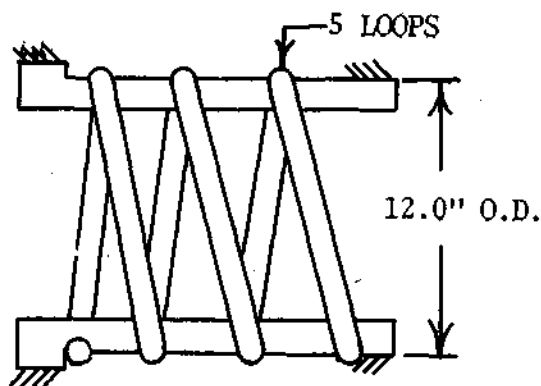
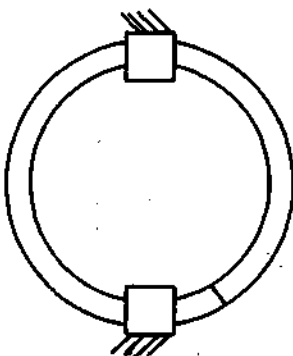
1. The maximum possible rate of change in Freon temperature at the heat exchanger inlet, and
2. The effect of Freon temperature changes at the heat exchanger outlet on other elements in the system, which determine control specifications.

A - Thermal Stresses - Minimal

B - Vibration Stresses -

Requirement = 24 G's + 10^7 cycles equivalent sine

Mounted as shown with full weight (1) assumed to be supported by outer tube (1.0 O.D., .028" wall, AA5062) gives factor of safety >1.5 for 10^7 cycles.



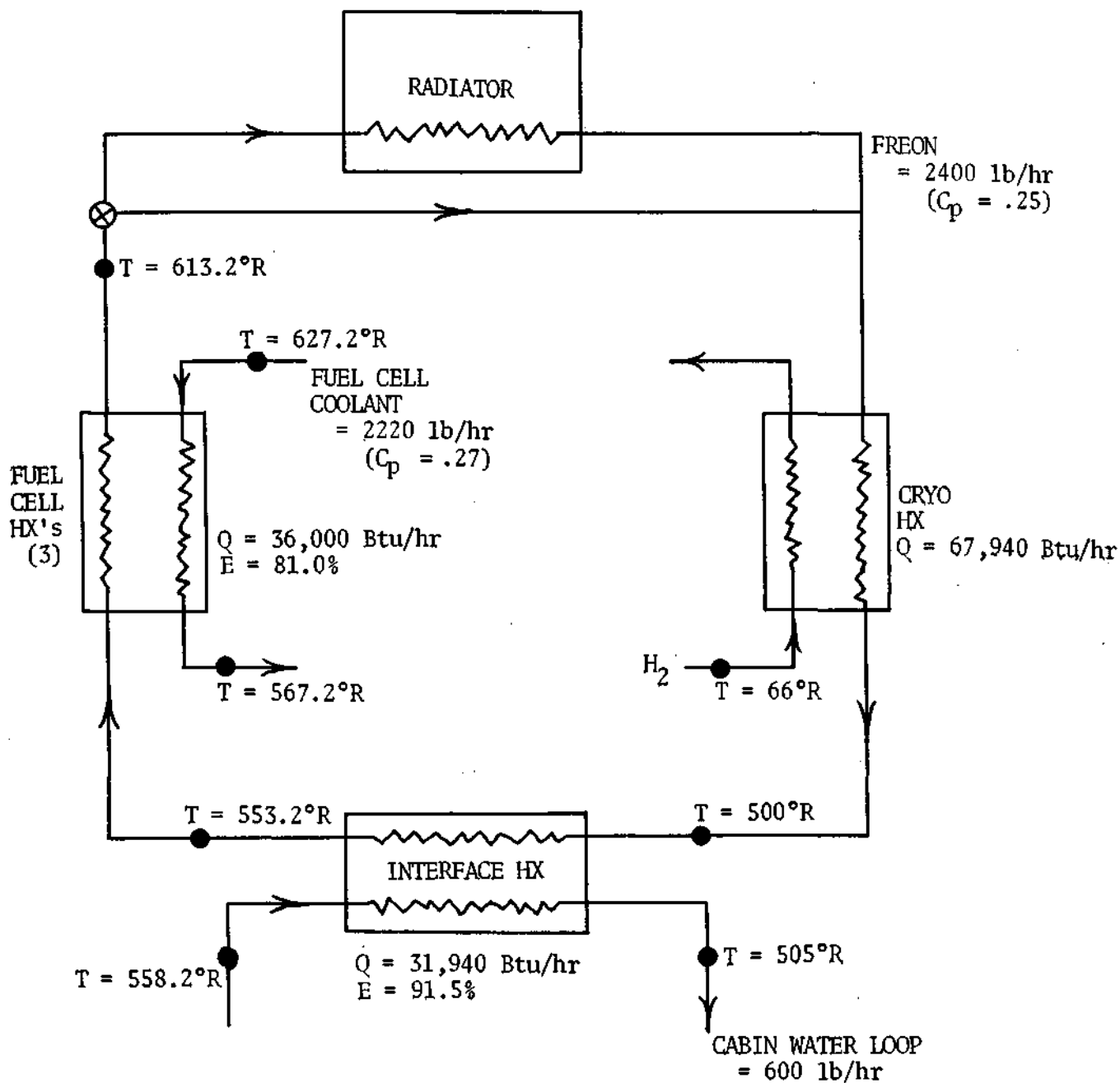
(1) Weight equal to 20 lbs., including insulation.

C - Shock - not a determining factor requirement on 20 G's.

CONCLUSION - There are no foreseeable structural problems in the design.

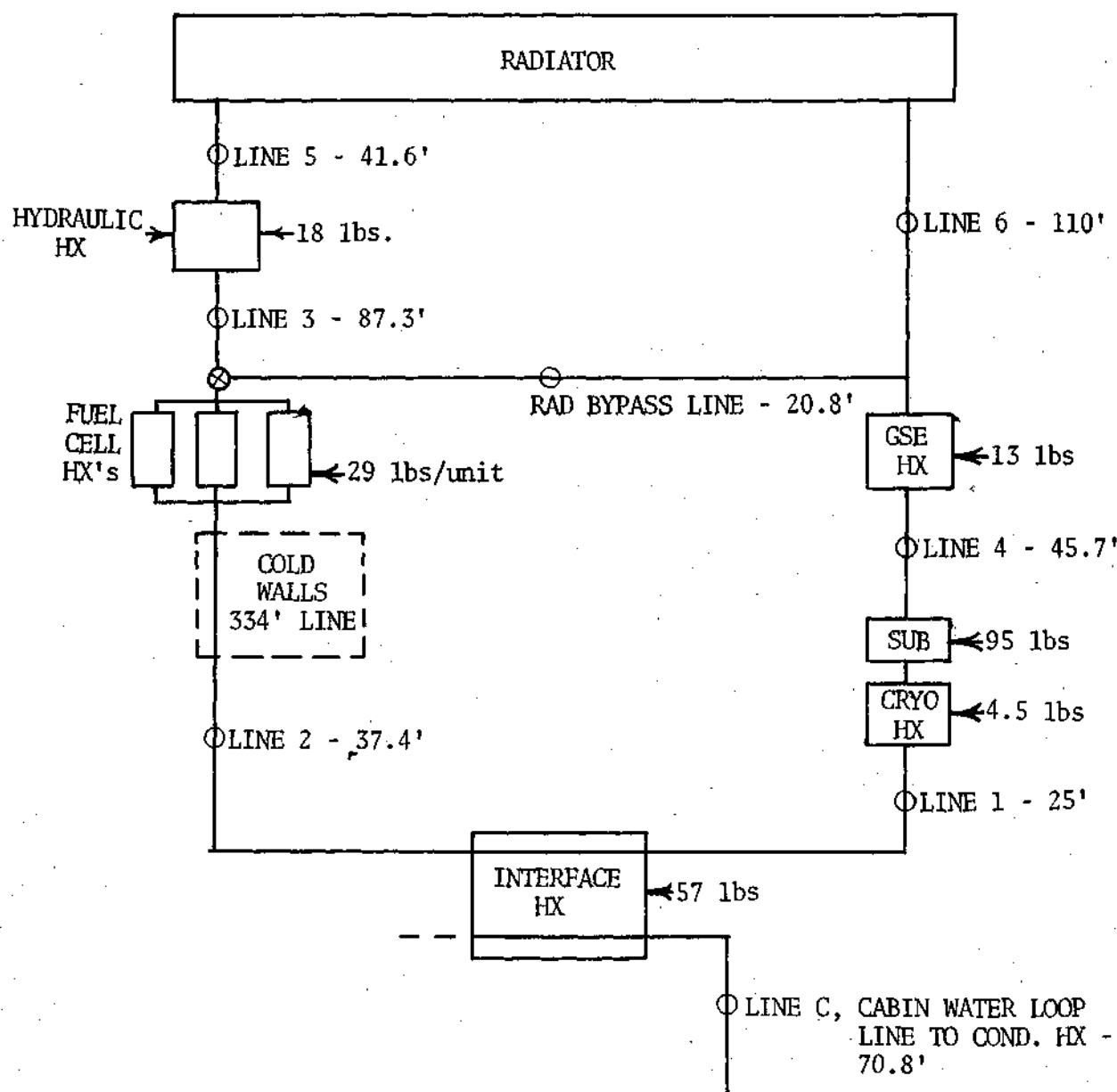
STRESS SUMMARY - CRYOGENIC H₂ HEAT EXCHANGER

FIGURE 3



FREON LOOP TEMPERATURES AND HEAT LOADS
AT ASSUMED DESIGN POINT

FIGURE 4



FREON LOOP LINES - 1" O.D. X .028" wall aluminum
metal wt per ft = .102 lb

WATER LOOP LINES - 1" O.D. X .028" wall stainless st.
metal wt per ft = .300 lb

FREON LOOP LINE LENGTHS AND UNIT WEIGHTS

FIGURE 5

TABLE I
DYNAMIC PARAMETERS

	NTU*	FREON TRANSPORT TIME, Sec.	OTHER FLUID TRANSPORT TIME, Sec.	METAL HEAT *** CAPACITY, Btu/°R **	FREON HEAT *** CAPACITY, Btu/°R	OTHER FLUID HEAT *** CAPACITY, Btu/°R	FREON hA, Btu hr °R	OTHER FLUID hA, Btu hr °R
FREON LOOP LINES, PER FOOT	.0565	.606	-	.0218	.101	-	33.9	-
WATER LOOP LINES, PER FOOT	.0687	-	1.82	.0328	-	.304	-	41.2
INTERFACE HEAT EXCHANGER	10.60	20.5	61.9	11.25	3.43	10.3	6380.	6380.
FUEL CELL HEAT EXCHANGERS (3 IN PARALLEL)	4.28	23.4	11.2	4.41	1.31	1.87	857.	857.
GSE HEAT EXCHANGER	7.67	4.1	-	2.78	.68	-	4570	-
SUBLIMATOR	4.43	3.1	-	5.22	.52	-	2660	-

* NTU = hA/WC_p or UA/WC_p in active heat exchangers

** Half of metal plus stagnant fluid in active heat exchangers

***Heat Capacity = Item Weight x Specific Heat = $Q/\Delta T$

A maximum rate of change in Freon inlet temperature of 20°R per minute was assumed for normal operation, with possibly greater rates occurring during start-up if the radiator were suddenly by-passed. The control characteristic selected for stability during normal operation was a full range of valve travel in 200 seconds and a $\pm 1^{\circ}\text{R}$ dead band.

Under the present contract a mathematical model of the thermal dynamics of the Freon loop was prepared and programmed as an addition to the cryogenic heat exchanger computer program. Using the expanded computer program the requirements for the cryogenic heat exchanger control were reexamined, with particular attention to start-up conditions.

The Freon loop system modeled was based on information obtained from the North American Shuttle ECS baseline (March 1972) and on studies conducted by Hamilton Standard. The active elements in the Freon loop during operation of the cryogenic heat exchanger with heat flows, fluid flow rates and temperatures at the assumed design point are shown in figure 4. Line lengths and component weights are shown in figure 5. The model includes the Freon loop completed by the radiator bypass but not the radiator, hydraulic heat exchanger or associated lines. The line in the cabin water loop from the interface heat exchanger to the cabin condensing heat exchanger is also included in order to determine the effect of Freon loop temperature changes on the temperature of cooling water entering the condenser.

The mathematical model and computer program are discussed in the next section and the controls reevaluation in the following one.

Description of Math Model and Computer Program

The components in the Freon loop are one of two types from the point of view of thermal dynamics:

1. Those in which heat transfer occurs only between Freon and metal, i.e. transport lines and the inactive GSE heat exchanger and sublimator which have no second fluid flowing. Freon and metal in these components come to the same temperature at steady state.
2. Active heat exchangers (interface and fuel cell heat exchangers).

Component models for each type were selected for their simplicity and tolerance to input changes. Where required for accuracy, components were divided into two or more segments.

Nomenclature:

A	=	Heat transfer surface area
Cp	=	Specific heat
m	=	Mass
NTU	=	UA/W (Number of Thermal Units)
Q	=	Heat flow
T	=	Temperature
t	=	Time
Δt	=	Finite interval in time
U	=	Heat transfer coefficient (between two fluids in heat exchangers, between fluid and wall in lines)
W	=	Flow rate
E	=	Heat exchanger effectiveness
Θ_w	=	Wall time const = $(mCp)_{wall}/UA$
τ	=	Fluid transport time
e	=	Base Napierian Logarithm (2.7183)
ω	=	Frequency (radians)
j	=	$\sqrt{-1}$

Transport Lines and Inactive Heat Exchangers

The fluid temperature at the outlet of a line with a uniform tube wall temperature is:

$$T \text{ fluid out} = T \text{ fluid } e^{-NTU} + T \text{ wall } (1 - e^{-NTU})$$

The rate of change in average wall temperature, assuming no heat transfer from the outside of the tube, is:

$$\frac{d T_{wall}}{dt} = \frac{Q \text{ to wall}}{(mCp)_{wall}} = \frac{(W C_p)_{fluid} (T_{in} - T_{out})}{(mCp)_{wall}}$$

Employing the approximation that the assumed uniform wall temperature in the first equation changes as the average, these two equations may be combined to yield an equation for rate of change in fluid outlet temperature:

$$\frac{dT_{\text{out}}^{\text{fluid}}}{dt} = \frac{dT_{\text{in}}^{\text{fluid}}}{dt} e^{-NTU} + \frac{(WCp)_{\text{fluid}}}{(mCp)_{\text{wall}}} (1 - e^{-NTU}) (T_{\text{in}}^{\text{fluid}} - T_{\text{out}}^{\text{fluid}})$$

When using this equation, the fluid outlet temperature must be delayed by the time it takes a particle of fluid to pass through the line before it becomes the inlet temperature to the next element.

The approximation involved in this equation for rate of change in outlet temperature does not introduce significant error when the NTU is on the order of 2 or less, as may be confirmed by calculating frequency response using this equation and comparing it to frequency response calculated using the exact transfer function for insulated lines.

The above equation converted to a transfer function is:

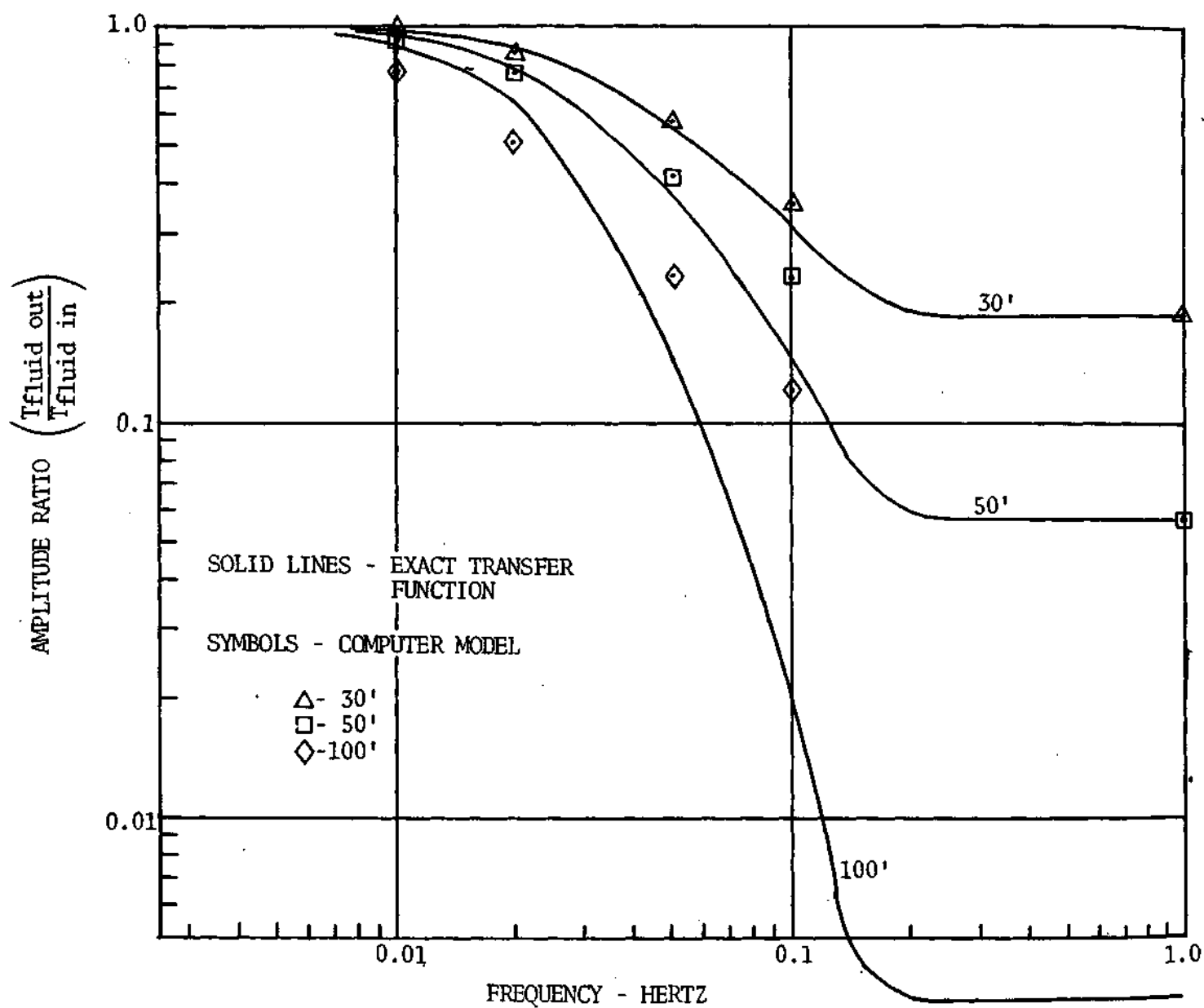
$$\frac{T_{\text{out}}^{\text{fluid}}}{T_{\text{in}}^{\text{fluid}}} = \frac{(j\omega e^{-NTU} + (1 - e^{-NTU}) \frac{(WCp)_{\text{fluid}}}{(mCp)_{\text{wall}}}) e^{-j\omega\tau}}{j\omega + (1 - e^{-NTU}) \frac{(WCp)_{\text{fluid}}}{(mCp)_{\text{wall}}}}$$

while the exact transfer function is:

$$\frac{T_{\text{out}}^{\text{fluid}}}{T_{\text{in}}^{\text{fluid}}} = e^{-\left[j\omega \left(\tau + \frac{NTU}{\omega^2 + \frac{1}{\Theta_w^2}} \right) + \frac{NTU \omega^2}{\omega^2 + \frac{1}{\Theta_w^2}} \right]}$$

Figure 6 shows a comparison of frequency response calculations using these two equations. Three lengths of Freon loop lines are shown. The NTU, which is proportional to line length, reaches a value of 2 at a line length about 36 feet. The approximate equation used in the computerized math model provides a reasonably accurate frequency response for a 30 foot line length, diverges somewhat from the exact solution at 50 feet, and becomes very inaccurate at a length of 100 feet.

The logic in modeling the inactive heat exchangers in the Freon loop (sublimator and GSE heat exchanger) is precisely the same as for the lines, since the only significant heat transfer occurs between Freon and heat exchanger metal. Due to the large heat transfer surface area in heat exchangers, however, the NTU is typically large. Consequently, it is



FREON LOOP LINE TEMPERATURE TRANSFER FUNCTIONS

FIGURE 6

necessary to break heat exchangers into segments having NTU's on the order of two. The sublimator was broken into two segments and the GSE heat exchanger into four. Of the transport lines in the model, only the water loop line from the interface heat exchanger to the condenser required more than one segment.

The differential equation for fluid leaving each line or inactive heat exchanger segment was put into the following form for solution by finite difference techniques using a computer:

$$T_{\text{out}}^{\text{fluid}}(1) = T_{\text{out}}^{\text{fluid}}(1) + \left(T_{\text{previous segment}}^{\text{fluid out}}(n) - T_{\text{out}}^{\text{fluid}}(1) \right) x$$

$$\frac{WC_p}{mC_p} \frac{\text{fluid}}{\text{wall}} (1 - e^{-NTU}) x \Delta t$$

$$+ \left(T_{\text{previous segment}}^{\text{fluid out}}(n-1) - T_{\text{previous segment}}^{\text{fluid out}}(n) \right) x e^{-NTU}$$

$$\text{where } n = \frac{\tau_{\text{segment}}}{\Delta t}$$

Values indexed n are values delayed by the fluid transport time in a segment, and are generated from currently calculated values (subscripted (1)) by setting:

$$T(n) = T(n-1)$$

for each of 2 through n values at each time step Δt .

Active Heat Exchangers

The steady state temperature of the cold fluid at the outlet of a counterflow heat exchanger, in terms of effectiveness, is:

$$T_{\text{out}}^{\text{cold}} = E \left(T_{\text{in}}^{\text{hot}} - T_{\text{in}}^{\text{cold}} \right) + T_{\text{in}}^{\text{cold}}$$

The steady state heat flow to the cold fluid is therefore:

$$Q_{s s} = WC_p \left(T_{\text{out}}^{\text{cold}} - T_{\text{in}}^{\text{cold}} \right) = WC_p E \left(T_{\text{in}}^{\text{hot}} - T_{\text{in}}^{\text{cold}} \right)$$

The excess of the instantaneous heat flow to the cold fluid over the steady state heat flow at the instantaneous inlet temperatures determines the rate of change of heat exchanger wall material (one half of which is assigned to each fluid):

$$-(mCp)_{\text{wall}} \frac{dT_{\text{wall}}}{dt} = WCp \left(T_{\text{out}}^{\text{cold}} - T_{\text{in}}^{\text{cold}} \right) - WCp E \left(T_{\text{in}}^{\text{hot}} - T_{\text{in}}^{\text{cold}} \right)$$

$$\frac{dT_{\text{wall}}}{dt} = \frac{(WCp)_{\text{cold}}}{(mCp)_{\text{wall}}} \left[(1-E) T_{\text{in}}^{\text{cold}} + E T_{\text{in}}^{\text{hot}} - T_{\text{out}}^{\text{cold}} \right]$$

Writing, as for the transport lines:

$$T_{\text{out}}^{\text{cold}} = T_{\text{in}}^{\text{cold}} e^{-NTU} + T_{\text{wall}} (1 - e^{-NTU})$$

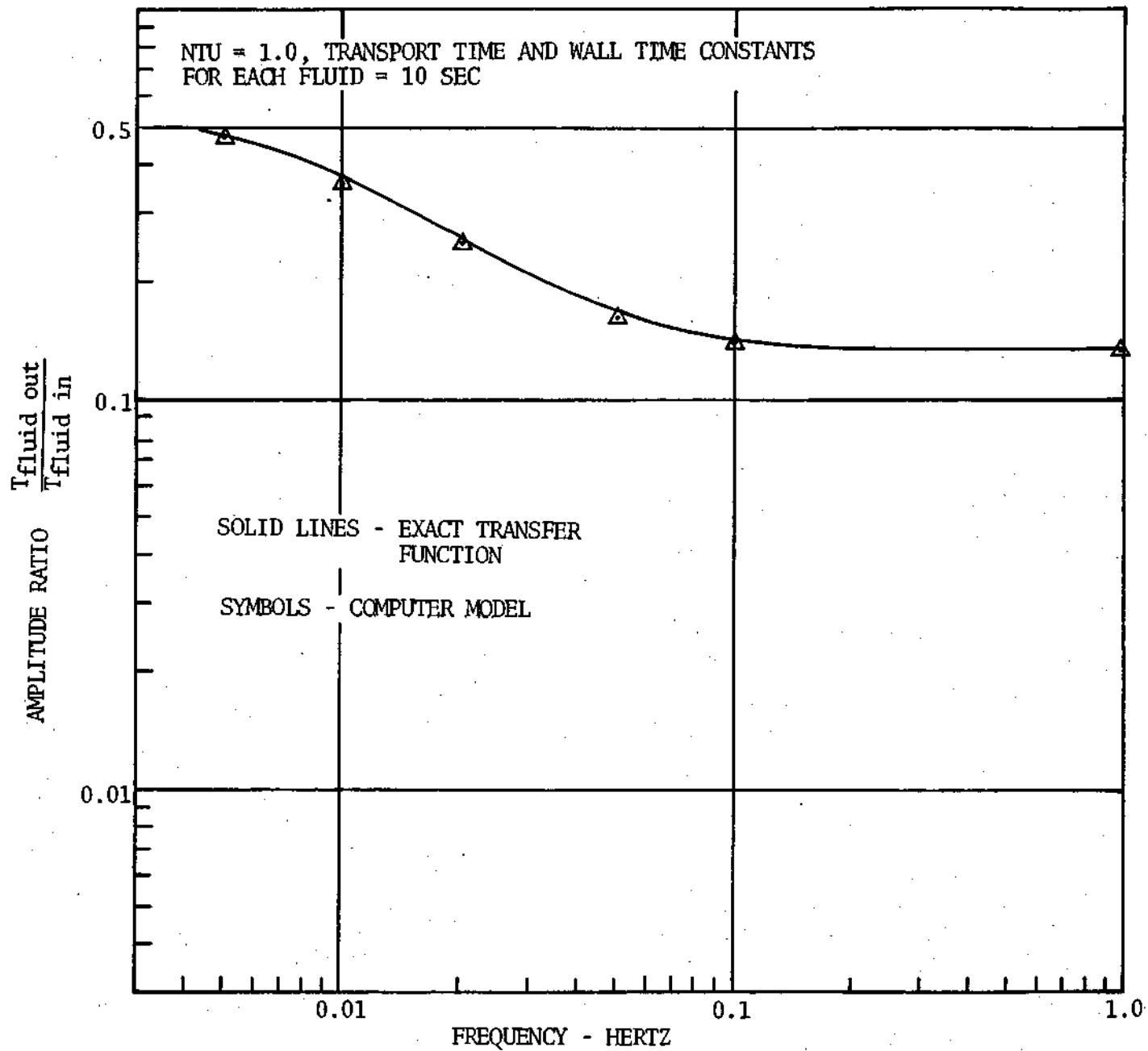
and combining this equation with the one above:

$$\frac{dT_{\text{out}}^{\text{cold}}}{dt} = \frac{dT_{\text{in}}^{\text{cold}}}{dt} e^{-NTU} + (1 - e^{-NTU}) \frac{(WCp)_{\text{cold}}}{(mCp)_{\text{wall}}} \left[(1-E) T_{\text{in}}^{\text{cold}} + E T_{\text{in}}^{\text{hot}} - T_{\text{out}}^{\text{cold}} \right]$$

An equation for the rate of change of the hot fluid temperature leaving the heat exchanger is similarly developed. As with the equation for transport lines, the accuracy of this equation may be tested by comparing calculated frequency response to that calculated using an exact counterflow heat exchanger transfer function. An exact temperature transfer function was derived by Gyftopolous and Smets, and presented by Ball in "Approximate Models for Distributed-Parameter Heat Transfer Systems", ISA Transactions, Vol. 3, No. 1. A straight through transfer function for a heat exchanger with an NTU of 1, and fluid transport times and wall time constants equal to 10 seconds for both fluids has been replotted from Ball's paper in figures 7 and 8.

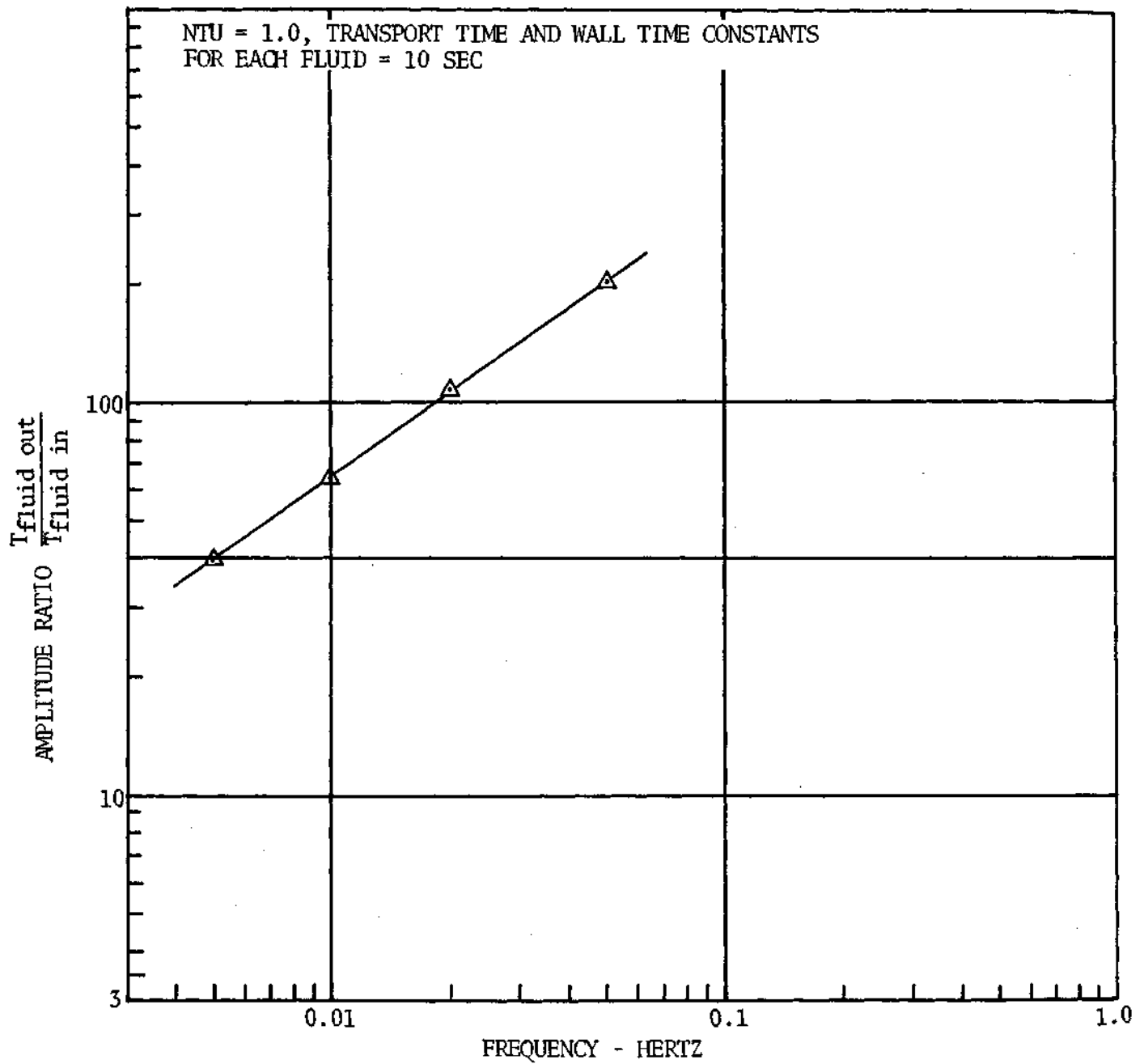
The straight through transfer function resulting from the approximate equation is:

$$\frac{T_{\text{out}}^{\text{cold}}}{T_{\text{in}}^{\text{cold}}} = \frac{\left(j\omega e^{-NTU} + (1-E) \frac{(WCp)_{\text{cold}}}{(mCp)_{\text{wall}}} (1 - e^{-NTU}) \right) e^{-j\omega\tau}}{j\omega + \frac{(WCp)_{\text{cold}}}{(mCp)_{\text{wall}}} (1 - e^{-NTU})}$$



HEAT EXCHANGER TEMPERATURE TRANSFER FUNCTIONS
AMPLITUDE RATIO VERSUS FREQUENCY

FIGURE 7



HEAT EXCHANGER TEMPERATURE TRANSFER FUNCTIONS
PHASE LAG VERSUS FREQUENCY

FIGURE 8

Values calculated using this equation are plotted as symbols in figures 7 and 8 and show good agreement with the exact frequency response. The accuracy decreases as NTU becomes large, and high effectiveness heat exchangers were broken into several segments (the interface heat exchanger five, and the fuel cell heat exchangers two each).

The finite difference form of the equation is:

$$T_{out}^{cold}(1) = T_{out}^{cold}(1) + \frac{(WCp)_{cold}}{(mCp)_{wall}} (1 - e^{-NTU}) \times$$

$$\left[E T_{in}^{hot}(1) + (1-E) T_{previous\ segment}^{cold, out}(n) - T_{out}^{cold}(1) \right] \times \Delta t$$

$$+ e^{-NTU} \times \left(T_{previous\ segment}^{cold, out}(n-1) - T_{previous\ segment}^{cold, out}(n) \right)$$

and $T_{(n)} = T_{(n+1)}$ at each time step.

Computer Program

A listing of the computer program including the cryogenic heat exchanger and its controls, and the thermal dynamics of the Freon loop is given below. The input is defined by comment cards at the beginning of the listing. Values for the dynamic parameters in the Freon loop used as input to the computer program in this study are summarized in Table I.

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C   DECK 505 TUBE IN TUBE COUNTERFLOW HEAT EXCHANGER WITH SUPERCRITICAL
C   H2 ENTERING INNER TUBE, FREON                               IN ANNULUS. CONTROL
C   OF FREON OUTLET TEMPERATURE BY MODULATING H2 FLOW USING CONSTANT
C   SPEED REVERSIBLE MOTOR TO ACTUATE INLET VALVE
C   SHUTLE FREON LOOP THERMAL DYNAMICS ADDED
C
C   INPUT IS LOADED BY STANDARD LOAD ROUTINE IN THE FOLLOWING LOCATIONS
C   GEOMETRICAL FACTORS
C   1   DI   INNER DIAMETER OF INNER TUBE, INCHES
C   2   TI   THICKNESS OF INNER TUBE WALL, INCHES
C   3   DO   INNER DIAMETER OF OUTER TUBE, INCHES
C   4   TO   THICKNESS OF OUTER TUBE WALL, INCHES
C   5   XL   HEAT EXCHANGER LENGTH, FEET
C   6   CD   COIL DIAMETER, INCHES
C   OPERATING CONDITIONS
C   7   WD   DESIGN VALUE OF H2 FLOW RATE, LB/HR
C   8   WI   INITIAL VALUE OF H2 FLOW RATE, LBS/HR
C   9   WF   FREON FLOW RATE, LB/HR
C   10  PD   H2 PRESSURE AT HEAT EXCHANGER INLET, PSIA
C   11  ZNPT1 NUMBER OF H2 INLET TEMPERATURES VS TIME POINTS INPUT
C   31-40 T1TIME  TIME AT WHICH H2 INLET TEMP IS DESIGNATED, 1 THRU 10, SEC
C   41-50 T1TEMP  H2 INLET TEMP AT TIME POINT 1 THRU 10, DEG R
C   MATERIAL PROPERTIES
C   13  CPF   FREON SPECIFIC HEAT, BTU/LB-DEG R
C   14  CONDF FREON THERMAL CONDUCTIVITY, BTU/HR-FT-DEG R
C   15  RHOF  FREON DENSITY, LB/FT3
C   16  VISF  FREON VISCOSITY, LB/HR-FT
C   17  CPW   TUBE METAL SPECIFIC HEAT, BTU/LB-DEG R
C   18  RHOW  TUBE METAL DENSITY, LB/FT3
C   19  XKW   TUBE METAL THERMAL CONDUCTIVITY, BTU/HR-FT-DEG R
C   71-86 T3TEMP  18 INPUT TEMPERATURES FOR H2 ENTHALPY, DEG R
C   91-106 T3ENTH  H2 ENTHALPY AT DESIGN PRESSURE, BTU/LB
C   CONTROL VARIABLES
C   20  GR   MOTOR RATE, PERCENT VALVE TRAVEL PER SECOND
C   21  SP   FREON OUTLET TEMPERATURE CONTROL SET POINT, DEG R
C   22  DBAND TEMP BAND AROUND SET POINT IN WHICH MOTOR IS INACTIVE, DEG R
C   23  ACDL  DELAY IN VALVE ACTUATOR FROM TIME OF TC SIGNAL, SEC
C   24  TCTC  THERMOCOUPLE TIME CONSTANT, SEC
C   COMPUTATIONAL CONSTANTS
C   25  TMAX  TIME AT WHICH TRANSIENT CALCULATION ENDS, SEC
C   26  ZN   NUMBER OF NODES IN HEAT EXCHANGER MODEL
C   FREON LOOP DYNAMICS
C   111 FLHAF  FREON LINE HA PER FOOT, BTU/(HR-FT-DEG R)
C   112 FLMCF  FREON LINE HEAT CAPACITY PER FOOT, BTU/(FT-DEG R)
C   113 FLMCLF FREON HEAT CAPACITY IN FOOT OF LINE, BTU/(FT-DEG R)
C   114 FL1L   LENGTH OF FREON LINE 1, FT
C   115 FL2L   LENGTH OF FREON LINE 2, FT
C   116 FL4L   LENGTH OF FREON LINE 4, FT
C   117 FBPL   LENGTH OF FREON RADIATOR BYPASS, FT
C   118 STTRL  LENGTH OF STAR TRACKER LINE, FT
C   121 WLHAF  WATER LINE HA PER FOOT, BTU/(HR-FT-DEG R)
C   122 WLMCF  WATER LINE HEAT CAPACITY PER FOOT, BTU/(FT-DEG R)
C   123 WLMCLF WATER HEAT CAPACITY IN FOOT OF LINE, BTU/(FT-DEG R)
C   124 WLCL   LENGTH OF WATER LINE FROM INTER HX TO CONDENSOR, FT
C   129 EIHx  EFFECTIVENESS OF INTERFACE HX
C   130 FIMCW  WALL HEAT CAPACITY FOR FREON IN INTERFACE HX, BTU/DEG R

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C 131 WIMCW WALL HEAT CAPACITY FOR WATER IN INTERFACE HX,BTU/DEG R
C 132 TAUIF TRANSPORT TIME OF FREON IN INTER HX, SEC
C 133 TAUIW TRANSPORT TIME OF WATER IN INTER HX, SEC
C 134 EFC EFFECTIVENESS OF FUEL CELL HXS
C 135 FCFMCW WALL HEAT CAP FOR FREON IN FUEL CELL HXS,BTU/DEG R
C 136 FC4MCW WALL HEAT CAP FOR FC43 IN FUEL CELL HXS,BTU/DEG R
C 137 TAUFCF TRANSPORT TIME OF FREON IN FUEL CELL HXS, SEC
C 138 TAUFC4 TRANSPORT TIME OF FC 43 IN FUEL CELL HXS, SEC
C 139 GSENTU NTU OF GSE HX,
C 140 GSEMCW HEAT CAPACITY OF GSE HX, BTU/DEG R
C 141 TAUGSE TRANSPORT TIME OF FREON IN GSE HX, SEC
C 142 SUBNTU NTU OF FREON IN SUBLIMATOR
C 143 SUBMCW HEAT CAPACITY OF SUBLIMATOR, BTU/DEG R
C 144 TAUSUB TRANSPORT TIME OF FREON IN SUBLIMATOR, SEC
C 150 TIHXW INITIAL TEMPERATURE OF WATER INTO INTER HX, DEG R
C 151 TFCWA INITIAL TEMPERATURE OF FC 43 INTO FUEL CELL HX A,DEG R
C 152 TFCWBI INITIAL TEMPERATURE OF FC 43 INTO FUEL CELL HX B,DEG R
C 153 TFCWCI INITIAL TEMPERATURE OF FC 43 INTO FUEL CELL HX C,DEG R
C 154 TBPI INITIAL TEMPERATURE OF RADIATOR BYPASS LINE,DEG R
C 155 TGSEI INITIAL TEMPERATURE OF GSE HX, DEG R
C 156 WW FLOW RATE IN WATER LOOP, LB/HR
```

```
DIMENSION PLT(502,13)
DIMENSION T(41), TF(41), TW(41), TS(41), ENT(41)
DIMENSION D(170), T1TEMP(10), T1TIME(10), T3ENTH(20), T3TEMP(20)
DIMENSION TLIN1(91),TLIN2(101),TLIN4(151),TBP(71),TLINC(501)
DIMENSION TSTTR(121),TIHXFC(21),TIHXWI(81),TSU1(21),TSUB(21)
DIMENSION TIHXFI(21),TIHXF1(21),TIHXF2(21),TIHXF3(21),TIHXF4(21)
DIMENSION TIHXW1(81),TIHXW2(81),TIHXW3(81),TIHXW4(81),TIHXWO(81)
DIMENSION TFCFA1(51),TFCFB1(51),TFCFC1(51),TFCFA0(51),TFCFB0(51)
DIMENSION TFCFC0(51),TFCWA1(31),TFCWA1(31),TFCWBI(31),TFCWCI(31)
DIMENSION TGS1(25), TGS2(25), TGS3(25), TGSE(25)
```

EQUIVALENCE

```
X (D( 1),DI ), (D( 2),TI ), (D( 3),DO ), (D( 4),TO ),
X (D( 5),XL ), (D( 6),CD ), (D( 7),WD ), (D( 8),WI ),
X (D( 9),WF ), (D(10),PD ), (D(11),ZNPT1 ), (D(12),ZNPT2 ),
X (D(13),CPF ), (D(14),CONDF ), (D(15),RHOF ), (D(16),VISF ),
X (D(17),CPW ), (D(18),RHOW ), (D(19),XKW ), (D(20),GR ),
X (D(21),SP ), (D(22),DBAND ), (D(23),ACDL ), (D(24),TCTC ),
X (D(25),TMAX ),(D(26),ZN ),(D(27),PLOT ),
X (D(31),T1TIME(1)), (D(41),T1TEMP(1)),
X (D(71),T3TEMP(1)), (D(91),T3ENTH(1)),
X (D(111),FLHAF ),(D(112),FLMCWF),(D(113),FLMCLF),(D(114),FL1L ),
X (D(115),FL2L ),(D(116),FL4L ),(D(117),FBPL ),(D(118),STTRL ),
X (D(121),WLHAF ),(D(122),WLMCWF),(D(123),WLMCLF),(D(124),WLCL ),
X (D(129),EIHx ),(D(130),FIMCW),(D(131),WIMCW),(D(132),TAUIF ),
X (D(133),TAUIW ),(D(134),EFC ),(D(135),FCFMCW),(D(136),FC4MCW),
X (D(137),TAUFCF),(D(138),TAUFC4),(D(139),GSENTU),(D(140),GSEMCW),
X (D(141),TAUGSE),(D(142),SUBNTU),(D(143),SUBMCW),(D(144),TAUSUB),
X (D(150),TIHXW ),(D(151),TFCWA ),(D(152),TFCWBI),(D(153),TFCWCI),
X (D(154),TBPI ),(D(155),TGSEI ),(D(156),WW )
```

DO 12 I=1,170

12 D(I)=0.

15 CONTINUE

CALL START

CALL LOAD(D)

WRITE(6,999) DI, TI, DC, TC, XL, CD, WD, WI, WF, PD, CPF, CONDF,

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X RHOF, VISE, CPW, RHOW, XKW, GR, SP, DBAND, ACDL, TCTC, FLHAF,
 X FLMCWF, FLMCLF, FL1L, FL2L, FL4L, FBPL, STTRL, WLHAF, WLMCWF, WLMCLF, WLCL,
 X EHX, FIMCW, WIMCW, TAUIF, TAUW, EFC, FCFMCW, FC4MCW, TAUFCF,
 X TAUFC4, GSENTU, GSEMCW, TAUGSE, SUBNTU, SUBMCW, TAUSUB, WW

999 FORMAT('O INPUT@/2X,

XG12.5,' DI @,G12.5,' TI @,G12.5,' DO @,G12.5,' TO @,
 XG12.5,' XL @,G12.5,' CD @/2X,
 XG12.5,' WD @,G12.5,' WI @,G12.5,' WF @,G12.5,' PD @,
 XG12.5,' CPF @,G12.5,' CONDF @/2X,
 XG12.5,' RHOF @,G12.5,' VISE @,G12.5,' CPW @,G12.5,' RHOW @,
 XG12.5,' XKW @,G12.5,' GR @/2X,
 XG12.5,' SP @,G12.5,' DBAND @,G12.5,' ACDL @,G12.5,' TCTC @,
 XG12.5,' FLHAF @,G12.5,' FLMCWF@/2X,
 XG12.5,' FLMCLF@,G12.5,' FL1L @,G12.5,' FL2L @,G12.5,' FL4L @,
 XG12.5,' FBPL @,G12.5,' STTRL @/2X
 XG12.5,' WLHAF @,G12.5,' WLMCWF@,G12.5,' WLMCLF@,G12.5,' WLCL @,
 XG12.5,' EHX @,G12.5,' FIMCW @/2X
 XG12.5,' WIMCW @,G12.5,' TAUIF @,G12.5,' TAUW@,G12.5,' EFC @,
 XG12.5,' FCFMCW@,G12.5,' FC4MCW@/2X
 XG12.5,' TAUFCF@,G12.5,' TAUFC4@,G12.5,' GSENTU@,G12.5,' GSEMCW@,
 XG12.5,' TAUGSE@,G12.5,' SUBNTU@/2X
 XG12.5,' SUBMCW@,G12.5,' TAUSUB@,G12.5,' WW @//)

N=ZN

IN1=N+1

ITER=1

ILP=0

IPI=PLOT

IPL=IPL+1

NPT1=ZNPT1

VPOS=0.

DVPOSA=0.0

TOFF=0.0

W=0.

TIME=0.0

TF(IN1)=TGSEI

ODI=DI+2*TI

A=.005454*DI*DI

AF=.005454*(DO*DO-ODI*ODI)

AS=.2618*DI*XL/N

ASF=AS*ODI/DI

ASS=.2618*DO*XL/N

DH=DI/12

DHF=(DO-ODI)/12

XMF=RHOF*AF*XL/N

XMW=RHOF*.005454*(ODI*ODI-DI*DI)*XL/N

XMS=RHOF*ASS*TO/12

DELT=XMF/WF*3600.

TAUFLF=FLMCLF/(WF*CPF)*3600.

FLNTUF=FLHAF/(WF*CPF)

WLNTUF=WLHAF/WW

ENTUL1=EXP(-FLNTUF*FL1L)

ENTUIH=EXP(-EIH/((1-EIH)*5))

ENTUL2=EXP(-FLNTUF*FL2L)

ENTUST=EXP(-FLNTUF*STTRL)

ENTUFC=(-EFC/((1-EFC)*2))

ENTUBP=EXP(-FLNTUF*FBPL)

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ENTUL4=EXP(-FLNTUF*FL4L)
ENTUGS=EXP(-GSENTU/4)
ENTUSU=EXP(-SURNTU/2)
ENTULC=EXP(-WLNTUF*WLCL)
CLCC=FLMCWF*3600./(WF*CPF)
CLIN1=CLCC*FL1L/(1-ENTUL1)
CLIN2=CLCC*FL2L/(1-ENTUL2)
CSTTR=CLCC*STTRL/(1-ENTUST)
CBP=CLCC*FBPL/(1-ENTUBP)
CLIN4=CLCC*FL4L/(1-ENTUL4)
CLINC=WLMCWF*WLCL/(WW*(1-ENTULC))*3600.
CIHXF=FLMCW/(WF*CPF*(1-ENTUIH)*5)*3600.
CIHXW=WIMCW/(WW*(1-ENTUIH)*5)*3600.
CFCF=FCFCW/(WF*CPF*(1-ENTUFC)*2)*3600.
CFC4=FC4MCW/(WW*(1-ENTUFC)*2)*3600.
CGSE=GSEMCW/(4*WF*CPF*(1-ENTUGS))*3600.
CSUR=SUBMCW/(2*WF*CPF*(1-ENTUSU))*3600.
NIHXF=TAUIF/(5*DELT)
NIHXW=TAUIW/(5*DELT)
NFCF=TAUFCF/(DELT*2)
NFC4=TAUFC4/(DELT*2)
NLIN1=TAUFLF*FL1L/DELT+NIHXF
NLIN2=TAUFLF*FL2L/DELT
NSTTR=TAUFLF*STTRL/DELT+NFCF
NBP=TAUFLF*FBPL/DELT
NLIN4=TAUFLF*FL4L/DELT
NLINC=WLMCLF*WLCL/WW*3600./DELT
NGSE=TAUGSE/(4*DELT)
NSUR=TAUSUB/(2*DELT)
TCEX=EXP(DELT/TCTC)
WTDRY=(XMW + XMS)*N
WTWET=WTDRY + XMF*N
RW2=(ODI-DI)/(48.*XKW)
BFA=1.+3.5*(DO-ODI)/CD
BFT=1.+3.5*DI/CD
PRF=VISF*CPF/CONDF
REF=DHF*WF/(AF*VISF)
  IF(REF-2300.)1,1,2
1  FFF=64./REF
  HF=4.86*CONDF/DHF
  GO TO 3
2  FFF=.184/(REF**.2)
  HF=.023*CONDF/DHF*REF**.8*PRF**.4*BFA
3  DPF=FFF*XL/DHF*(WF/AF)**2/(1.2E+11*RHOF)*BFA
  UFW=1/(1/HF+RW2)
  TCWF=XMW*CPW/(UFW*ASF)*3600.
  TCSE=XMS*CPW/(UFW*ASS)*3600.
  TCFW=XMF*CPF/(UFW*ASF)*3600.
  TCFS=XMF*CPF/(UFW*ASS)*3600.
  IF(DELT-TCWF)85,85,84
84  WRITE(6,1000) TCWF, DELT
1000 FORMAT('0*** N IS TOO SMALL FOR STABLE SOLUTION ***a/5x,
  X' TCWF#a,G13.5,10X,'DELT#a,G13.5)
  GO TO 15
85  TOUT=550.
  TFOUT=500.

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CALL UNLIN(NPT1,T1TIME,T1TEMP,TIME,T(1),IBUG)
CALL UNINT(18 ,T3TEMP,T3ENTH,T(1),ENT(1),IBUG)
DO 9 I=1,N
  T(I+1)=T(I)+(TOUT-T(1))/N
  TF(I)=TF(IN1)-(TF(IN1)-TFOUT)*(N-I+1)/N
  TW(I)=(TF(I)+T(I+1))/2
9   TS(I)=TF(I)
10  TFI=TF(1)
    TIT=T(IN1)
    GO TO 21
20  TF1=TF(1)
    TFN1=TF(IN1)
    CALL UNLIN(NPT1,T1TIME,T1TEMP,TIME,T(1),IBUG)
    CALL UNINT(18 ,T3TEMP,T3ENTH,T(1),ENT(1),IBUG)
21  OPTOT=0.0
    DO 100 I=1,N
      TFAB=(TF(I)+TF(I+1))/2
      IF(W-0.)30,30,31
30   DTW=(TFAB-TW(I))/TCWF
      GO TO 32
31   TAB=(T(I)+T(I+1))/2
      TFI1=(TAB+TW(I))/2
      IF(TFI1-300.)41,41,40
40   CPFIL=3.55
      COND=.00196*TFIL**.637
      GO TO 42
41   CPFIL=.1805*TFIL**.5223
      COND=.00021*TFIL**1.028
42   VIS=.000268*TFIL**.698
      PR=VIS*CPFIL/COND
      RE=DH*W/(A*VIS)*TAB/TFIL
      H=.023*COND/DH*RE**.8*PR**.4*8FT
      UW=1/(1/H+RW2)
      TCWH=XMW*CPW/(UW*AS)*3600.
      DENT=UW*AS/W*(TW(I)-TAB)
      ENT(I+1)=ENT(I)+DENT
      CALL UNINT(18 ,T3ENTH,T3TEMP,ENT(I+1),T(I+1),IBUG)
      TAB=(T(I)+T(I+1))/2
      FF=.184/(RE **.2)
      RHO=PD*144./(55.1*TAB)
      DP=FF*XL/(N*DH)*(W/A)**2/(1.2E+11*RHO)*8FT
      OPTOT=OPTOT+DP
      DTW=(TFAB-TW(I))/TCWF+(TAB-TW(I))/TCWH
32  TW(I)=DTW*DELT/2+TW(I)
      DTS=(TFAB-TS(I))/TCSE
      TS(I)=DTS*DELT/2+TS(I)
      DTF=(TW(I)-TFAB)/TCFW+(TS(I)-TFAB)/TCFS
      TF(I)=DTF*DELT+TF(I+1)
      IF(W-0.)33,33,34
33  DTW=(TFAB-TW(I))/TCWF
      GO TO 35
34  DTW=(TFAB-TW(I))/TCWF+(TAB-TW(I))/TCWH
35  TW(I)=DTW*DELT/2+TW(I)
      DTS=(TFAB-TS(I))/TCSE
100 TS(I)=DTS*DELT/2+TS(I)
      IF(TIME-DELT/2)101,101,110

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101 IF(ABS(TFIT-TF(1))-.01)102,102,103
102 IF(ABS(TIT-T(IN1))-.01)104,104,103
103 ITER=1
    GO TO 10
104 ITER=ITER+1
    IF(ITER-N)10,10,105
105 TCC=TF(1)
    TF1=TF(1)
    DO 777 LF=1,NLIN1
777 TLIN1(LF)=TF(1)
    SEGIHX=EIHX*(TIHXW -TF(1))/5
    EIHX=EIHX/(EIHX+(1-EIHX)*5)
    DO 782 LF=1,NIHXF
    TIHXF1(LF)=TF(1)+SEGIHX*1
    TIHXF2(LF)=TF(1)+SEGIHX*2
    TIHXF3(LF)=TF(1)+SEGIHX*3
    TIHXF4(LF)=TF(1)+SEGIHX*4
782 TIHXF0(LF)=TF(1)+SEGIHX*5
    DO 787 LF=1,NIHXW
    TIHXL1(LF)=TIHXW
    TIHXL1(LF)=TIHXW -SEGIHX*4
    TIHXL2(LF)=TIHXW -SEGIHX*3
    TIHXL3(LF)=TIHXW -SEGIHX*2
    TIHXL4(LF)=TIHXW -SEGIHX
787 TIHXL0(LF)=TIHXW -SEGIHX*5
    DO 778 LF=1,NLIN2
778 TLIN2(LF)=TIHXL0(NIHXF)
    DO 786 LF=1,NSTTR
786 TSTTR(LF)=TLIN2(NLIN2)
    SEGFCB=EFC*(TFCWA-TLIN2(NLIN2))/2
    SEGFCB=EFC*(TFCWB1-TLIN2(NLIN2))/2
    EFC=EFC/(EFC+(1-EFC)*2)
    DO 783 LF=1,NFCF
    TFCFA1(LF)=TLIN2(NLIN2)+SEGFCB
    TFCFB1(LF)=TLIN2(NLIN2)+SEGFCB
    TFCFC1(LF)=TLIN2(NLIN2)+SEGFCB
    TFCFA0(LF)=TLIN2(NLIN2)+SEGFCB*2
    TFCFB0(LF)=TLIN2(NLIN2)+SEGFCB*2
783 TFCFC0(LF)=TLIN2(NLIN2)+SEGFCB*2
    TFCFO=(TFCFA0(NFCF)+TFCFB0(NFCF)+TFCFC0(NFCF))/3
    DO 789 LF=1,NFC4
    TFCWAI(LF)=TFCWA
    TFCWAI(LF)=TFCWA-SEGFCB
    TFCWB1(LF)=TFCWB1-SEGFCB
789 TFCWC1(LF)=TFCWC1-SEGFCB
    DO 779 LF=1,NBP
779 TBP1(LF)=TBP1
    DO 784 LF=1,NGSE
    TGS1(LF)=TGSEI
    TGS2(LF)=TGSEI
    TGS3(LF)=TGSEI
784 TGSE(LF)=TGSEI
    DO 780 LF=1,NLIN4
780 TLIN4(LF)=TGSEI
    DO 785 LF=1,NSUB
    TSU1(LF)=TGSEI

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785 TSUB(LF)=TGSEI
    DO 781 LF=1,NLINC
781 TLINC(LF)=TIHXWO(NIHXW)
    WRITE(6,1001) REF, HF, DPF, RE, H, DPTOT, WTDY, WTWT,
    X TCWF, TCWH, TCSF, TCFW
1001 FORMAT('OINITIAL CONDITIONS@/2X,
    XG12.5,' REF @,G12.5,' HF @,G12.5,' DPF @,G12.5,' RE @,
    XG12.5,' H @,G12.5,' DPTOT@/2X,
    XG12.5,' WTD @,G12.5,' WTW @,G12.5,' TCWF@,G12.5,' TCWH@,
    XG12.5,' TCSF @,G12.5,' TCFW @//)
    WRITE(6,1003)
1003 FORMAT('O H2 TEMPS, 1 THRU N@1@)
    WRITE(6,1002) (T(K),K=1,IN1)
1002 FORMAT(2X,10G12.5)
    WRITE(6,1004)
1004 FORMAT('OWALL TEMPS,1 THRU N@)
    WRITE(6,1002) (TW(K),K=1,N)
    WRITE(6,1005)
1005 FORMAT('O FREON TEMPS, 1 THRU N+1@)
    WRITE(6,1002) (TF(K),K=1,IN1)
110 TW1=TW(1)+UFW*RW2*(TF(1)-TW(1))
    DTLIN1=(TF(1)-TLIN1(1))/CLIN1
    TLIN1(1)=TLIN1(1)+DTLIN1*DELT+(TF(1)-TF1)*ENTUL1
    DO 888 LF=2,NLIN1
888 TLIN1(NLIN1-LF+2)=TLIN1(NLIN1-LF+1)
    DTIF1=(EIH*TIHXW1(1)+(1-EIH)*TLIN1(NLIN1)-TIHXF1(1))/CIHXF*DELT
    TIHXF1(1)=TIHXF1(1)+DTIF1+(TLIN1(NLIN1-1)-TLIN1(NLIN1))*ENTUIH
    DTIF2=(EIH*TIHXW2(1)+(1-EIH)*TIHXF1(NIHXF)-TIHXF2(1))/CIHXF*DELT
    TIHXF2(1)=TIHXF2(1)+DTIF2+(TIHXF1(NIHXF-1)-TIHXF1(NIHXF))*ENTUIH
    DTIF3=(EIH*TIHXW3(1)+(1-EIH)*TIHXF2(NIHXF)-TIHXF3(1))/CIHXF*DELT
    TIHXF3(1)=TIHXF3(1)+DTIF3+(TIHXF2(NIHXF-1)-TIHXF2(NIHXF))*ENTUIH
    DTIF4=(EIH*TIHXW4(1)+(1-EIH)*TIHXF3(NIHXF)-TIHXF4(1))/CIHXF*DELT
    TIHXF4(1)=TIHXF4(1)+DTIF4+(TIHXF3(NIHXF-1)-TIHXF3(NIHXF))*ENTUIH
    DTIF5=(EIH*TIHXW1(1)+(1-EIH)*TIHXF4(NIHXF)-TIHXF5(1))/CIHXF*DELT
    TIHXF5(1)=TIHXF5(1)+DTIF5+(TIHXF4(NIHXF-1)-TIHXF4(NIHXF))*ENTUIH
    DTIW5=(EIH*TIHXF4(1)+(1-EIH)*TIHXW1(NIHXW)-TIHXW4(1))/CIHXW*DELT
    TIHXW4(1)=TIHXW4(1)+DTIW5+(TIHXW1(NIHXW-1)-TIHXW1(NIHXW))*ENTUIH
    DTIW4=(EIH*TIHXF3(1)+(1-EIH)*TIHXW4(NIHXW)-TIHXW3(1))/CIHXW*DELT
    TIHXW3(1)=TIHXW3(1)+DTIW4+(TIHXW4(NIHXW-1)-TIHXW4(NIHXW))*ENTUIH
    DTIW3=(EIH*TIHXF2(1)+(1-EIH)*TIHXW3(NIHXW)-TIHXW2(1))/CIHXW*DELT
    TIHXW2(1)=TIHXW2(1)+DTIW3+(TIHXW3(NIHXW-1)-TIHXW3(NIHXW))*ENTUIH
    DTIW2=(EIH*TIHXF1(1)+(1-EIH)*TIHXW2(NIHXW)-TIHXW1(1))/CIHXW*DELT
    TIHXW1(1)=TIHXW1(1)+DTIW2+(TIHXW2(NIHXW-1)-TIHXW2(NIHXW))*ENTUIH
    DTIW1=(EIH*TLIN1(NLIN1-NIHXF)+(1-EIH)*TIHXW1(NIHXW)-TIHXWO(1))
    X /CIHXW*DELT
    TIHXWO(1)=TIHXWO(1)+DTIW1+(TIHXW1(NIHXW-1)-TIHXW1(NIHXW))*ENTUIH
    TIHXW1(1)=TIHXW
    DO 903 LF=2,NIHXF
    TIHXF1(NIHXF-LF+2)=TIHXF1(NIHXF-LF+1)
    TIHXF2(NIHXF-LF+2)=TIHXF2(NIHXF-LF+1)
    TIHXF3(NIHXF-LF+2)=TIHXF3(NIHXF-LF+1)
    TIHXF4(NIHXF-LF+2)=TIHXF4(NIHXF-LF+1)
903 TIHXF5(NIHXF-LF+2)=TIHXF5(NIHXF-LF+1)
    DO 911 LF=2,NIHXW
    TIHXW1(NIHXW-LF+2)=TIHXW1(NIHXW-LF+1)
    TIHXW4(NIHXW-LF+2)=TIHXW4(NIHXW-LF+1)

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TIHXW3(NIHXW-LF+2)=TIHXW3(NIHXW-LF+1)
TIHXW2(NIHXW-LF+2)=TIHXW2(NIHXW-LF+1)
TIHXW1(NIHXW-LF+2)=TIHXW1(NIHXW-LF+1)
911 TIHXW0(NIHXW-LF+2)=TIHXW0(NIHXW-LF+1)
DTL2DT=(TIHXFO(NIHXF)-TLIN2(1))/CLIN2*DELT
TLIN2(1)=TLIN2(1)+DTL2DT+(TIHXFO(NIHXF-1)-TIHXFO(NIHXF))*ENTUL2
DO 889 LF=2,NLIN2
889 TLIN2(NLIN2-LF+2)=TLIN2(NLIN2-LF+1)
DTSTTR=(TLIN2(NLIN2)-TSTTR(1))/CSTTR
TSTTR(1)=TSTTR(1)+DTSTTR*DELT+(TLIN2(NLIN2-1)-TLIN2(NLIN2))*ENTUST
DO 913 LF=2,NSTTR
913 TSTTR(NSTTR-LF+2)=TSTTR(NSTTR-LF+1)
DTFA1=(EFC*TFCWA1(1)+(1-EFC)*TSTTR(NSTTR)-TFCFA1(1))/CFCF*DELT
TFCFA1(1)=TFCFA1(1)+DTFA1+(TSTTR(NSTTR-1)-TSTTR(NSTTR))*ENTUFC
DTFB1=(EFC*TFCWB1(1)+(1-EFC)*TSTTR(NSTTR)-TFCFB1(1))/CFCF*DELT
TFCFB1(1)=TFCFB1(1)+DTFB1+(TSTTR(NSTTR-1)-TSTTR(NSTTR))*ENTUFC
DTFC1=(EFC*TFCWC1(1)+(1-EFC)*TSTTR(NSTTR)-TFCFC1(1))/CFCF*DELT
TFCFC1(1)=TFCFC1(1)+DTFC1+(TSTTR(NSTTR-1)-TSTTR(NSTTR))*ENTUFC
DTFA0=(EFC*TFCWA1(1)+(1-EFC)*TFCFA1(NFCF)-TFCFA0(1))/CFCF*DELT
TFCFA0(1)=TFCFA0(1)+DTFA0+(TFCFA1(NFCF-1)-TFCFA1(NFCF))*ENTUFC
DTFB0=(EFC*TFCWB1(1)+(1-EFC)*TFCFB1(NFCF)-TFCFB0(1))/CFCF*DELT
TFCFB0(1)=TFCFB0(1)+DTFB0+(TFCFB1(NFCF-1)-TFCFB1(NFCF))*ENTUFC
DTFC0=(EFC*TFCWC1(1)+(1-EFC)*TFCFC1(NFCF)-TFCFC0(1))/CFCF*DELT
TFCFC0(1)=TFCFC0(1)+DTFC0+(TFCFC1(NFCF-1)-TFCFC1(NFCF))*ENTUFC
DTWA1=(EFC*TFCWA1(1)+(1-EFC)*TFCWA1(NFC4)-TFCWA1(1))/CFC4*DELT
TFCWA1(1)=TFCWA1(1)+DTWA1+(TFCWA1(NFC4-1)-TFCWA1(NFC4))*ENTUFC
DTWB1=(EFC*TFCWB1(1)+(1-EFC)*TFCWB1(NFC4)-TFCWB1(1))/CFC4*DELT
TFCWB1(1)=TFCWB1(1)+DTWB1
DTWC1=(EFC*TFCWC1(1)+(1-EFC)*TFCWC1(NFC4)-TFCWC1(1))/CFC4*DELT
TFCWC1(1)=TFCWC1(1)+DTWC1
TFCWA1(1)=TFCWA1
TFCF00=TFCF0
TFCF0=(TFCFA0(NFCF)+TFCFB0(NFCF)+TFCFC0(NFCF))/3
DO 893 LF=2,NFCF
TFCFA1(NFCF-LF+2)=TFCFA1(NFCF-LF+1)
TFCFB1(NFCF-LF+2)=TFCFB1(NFCF-LF+1)
TFCFC1(NFCF-LF+2)=TFCFC1(NFCF-LF+1)
TFCFA0(NFCF-LF+2)=TFCFA0(NFCF-LF+1)
TFCFB0(NFCF-LF+2)=TFCFB0(NFCF-LF+1)
893 TFCFC0(NFCF-LF+2)=TFCFC0(NFCF-LF+1)
DO 910 LF=2,NFC4
TFCWA1(NFC4-LF+2)=TFCWA1(NFC4-LF+1)
TFCWA1(NFC4-LF+2)=TFCWA1(NFC4-LF+1)
TFCWB1(NFC4-LF+2)=TFCWB1(NFC4-LF+1)
910 TFCWC1(NFC4-LF+2)=TFCWC1(NFC4-LF+1)
DTBP=(TFCF0-TBP(1))/CBP
TBP(1)=TBP(1)+DTBP*DELT+(TFCF0-TFCF00)*ENTUBP
DO 890 LF=2,NBP
890 TBP(NBP-LF+2)=TBP(NBP-LF+1)
DTGS1=(TBP(NBP)-TGS1(1))/CGSE
TGS1(1)=TGS1(1)+DTGS1*DELT+(TBP(NBP-1)-TBP(NBP))*ENTUGS
DTGS2=(TGS1(NGSE)-TGS2(1))/CGSE
TGS2(1)=TGS2(1)+DTGS2*DELT+(TGS1(NGSE-1)-TGS1(NGSE))*ENTUGS
DTGS3=(TGS2(NGSE)-TGS3(1))/CGSE
TGS3(1)=TGS3(1)+DTGS3*DELT+(TGS2(NGSE-1)-TGS2(NGSE))*ENTUGS
DTGSE=(TGS3(NGSE)-TGSE(1))/CGSE
```


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TGSE(1)=TGSE(1)+DTGSE*DELT+(TGS3(NGSE-1)-TGS3(NGSE))*ENTUGS
DO 898 LF=2,NGSE
TGS1(NGSE-LF+2)=TGS1(NGSE-LF+1)
TGS2(NGSE-LF+2)=TGS2(NGSE-LF+1)
TGS3(NGSE-LF+2)=TGS3(NGSE-LF+1)
898 TGSE(NGSE-LF+2)=TGSE(NGSE-LF+1)
DTLIN4=(TGSE(NGSE)-TLIN4(1))/CLIN4
TLIN4(1)=TLIN4(1)+DTLIN4*DELT+(TGSE(NGSE-1)-TGSE(NGSE))*ENTUL4
DO 891 LF=2,NLIN4
891 TLIN4(NLIN4-LF+2)=TLIN4(NLIN4-LF+1)
DTSU1=(TLIN4(NLIN4)-TSU1(1))/CSUB
TSU1(1)=TSU1(1)+DTSU1*DELT+(TLIN4(NLIN4-1)-TLIN4(NLIN4))*ENTUSU
DTSUB=(TSU1(NSUB)-TSUB(1))/CSUB
TSUB(1)=TSUB(1)+DTSUB*DELT+(TSU1(NSUB-1)-TSU1(NSUB))*ENTUSU
DO 895 LF=2,NSUB
TSU1(NSUB-LF+2)=TSU1(NSUB-LF+1)
895 TSUB(NSUB-LF+2)=TSUB(NSUB-LF+1)
TF(IN1)=TSUB(NSUB)
DTLCDT=(TIHXWD(NIHGX)-TLINC(1))/CLINC*DELT
TLINC(1)=TLINC(1)+DTLCDT+(TIHXWD(NIHGX-1)-TIHXWD(NIHGX))*ENTULC
DO 892 LF=2,NLINC
892 TLINC(NLINC-LF+2)=TLINC(NLINC-LF+1)
IF(IPL-IPL)113,112,112
113 IPL=1
WRITE(6,1007)
1007 FORMAT('0          SECONDS          F2 FLOW          T(1)          T(N+1)
X          TF(1)          TF(NG1)          TW(1)          TWF1 @)
WRITE(6,1006)TIME, W, T(1), T(IN1), TF(1), TF(IN1), TW(1), TWF1
WRITE(6,1008)
1008 FORMAT('0          INT HX FREON SEG 1 TO 4
X          INT HX WATER SEGS 4 TO 1          @)
WRITE(6,1006)TIHXF1(NIHXF),TIHXF2(NIHXF),TIHXF3(NIHXF),
X TIHXF4(NIHXF),TIHXW4(NIHGX),TIHXW3(NIHGX),TIHXW2(NIHGX),
X TIHXW1(NIHGX)
WRITE(6,1006)TSU1(NSUB),TLIN4(NLIN4),TGSE(NGSE),TGS1(NGSE),
X TBP(NBP),TFCFA0(NFCF),TFCFA1(NFCF),TSTTR(NSTTR)
WRITE(6,1006)TFCWA1(NFC4),TFCWA1(NFC4),DTFA1,DTWA1,TFCFB1(NFCF),
X TFCWB1(NFC4),TFCFC0(NFCF)
ILP=ILP+1
IF(ILP.GE.501) GO TO 112
PLT(ILP,1)=TIME
PLT(ILP,2)=TLIN4(NLIN4)
PLT(ILP,3)=TF(IN1)
PLT(ILP,4)=W
PLT(ILP,5)=TF(1)
PLT(ILP,6)=TLIN1(NLIN1)
PLT(ILP,7)=TIHXFO(NIHXF)
PLT(ILP,8)=TSTTR(NSTTR)
PLT(ILP,9)=TFCFO
PLT(ILP,10)=TBP(NBP)
PLT(ILP,11)=TGSE(NGSE)
PLT(ILP,12)=TIHXWD(NIHGX)
PLT(ILP,13)=TLINC(NLINC)
112 IPL=IPL+1
1006 FORMAT(2X,8G15.5)
TT=(TF1+TF(1))/2
```

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TCC=TT+(TCC-TT)/TCEX
TIME=TIME+DELT
IF(TIME-TMAX)120,120,200
120 DVPOS=DVPOS
417 IF((TLIN4(NLIN4-71)-TLIN4(NLIN4-72))/DELT+.4)418,418,419
419 IF((TLIN4(NLIN4-71)-TLIN4(NLIN4-72))/DELT-.4)4171,421,421
4171 IF(TCC-SP-DBAND/2)4191,418,418
418 DVPOS=GR
GO TO 422
4191 IF(TCC-SP+DBAND/2)421,420,420
420 DVPOS=0.0
GO TO 422
421 DVPOS=-GR
422 IF(DVPOS-DVPOS)424,423,424
423 IF(TIME-TOFF-ACDL)426,426,425
424 TOFF=TIME
GO TO 426
425 DVPOSA=DVPOS
426 VPOS=VPOS+DVPOSA*DELT
IF(VPOS)427,427,428
427 VPOS=0.0
GO TO 430
428 IF(VPOS-1.2)430,430,429
429 VPOS=1.2
430 W=WD/.966*SIN(1.57*VPOS/1.2)
GO TO 20
200 CONTINUE
IF(ILP .GE. 501) ILP=500
CALL CALULM(PLT,ILP,12,502)
GO TO 15
END

```

Controls Reevaluation

The H₂ feed control selected for the cryogenic heat exchanger in a study conducted under Contract NAS 9-12208 had the characteristics indicated in figure 9: a constant valve stroke travel rate equal to 200 seconds for full valve travel to the design flow rate, and a two degree dead band around the control set point. The H₂ source considered for the cryogenic heat exchanger was a supercritical tank for fuel cells. H₂ is tapped off an external pressurization system for this tank as shown in figure 10. The Freon temperature leaving the cryogenic heat exchanger is sensed, and the control is activated when the sensed temperature is one degree more or less than the 500°R set point.

Conservative non-ideal control characteristics assumed for the controls study were:

Thermocouple lag = 1.0 second

Actuator delay = 0.2 second

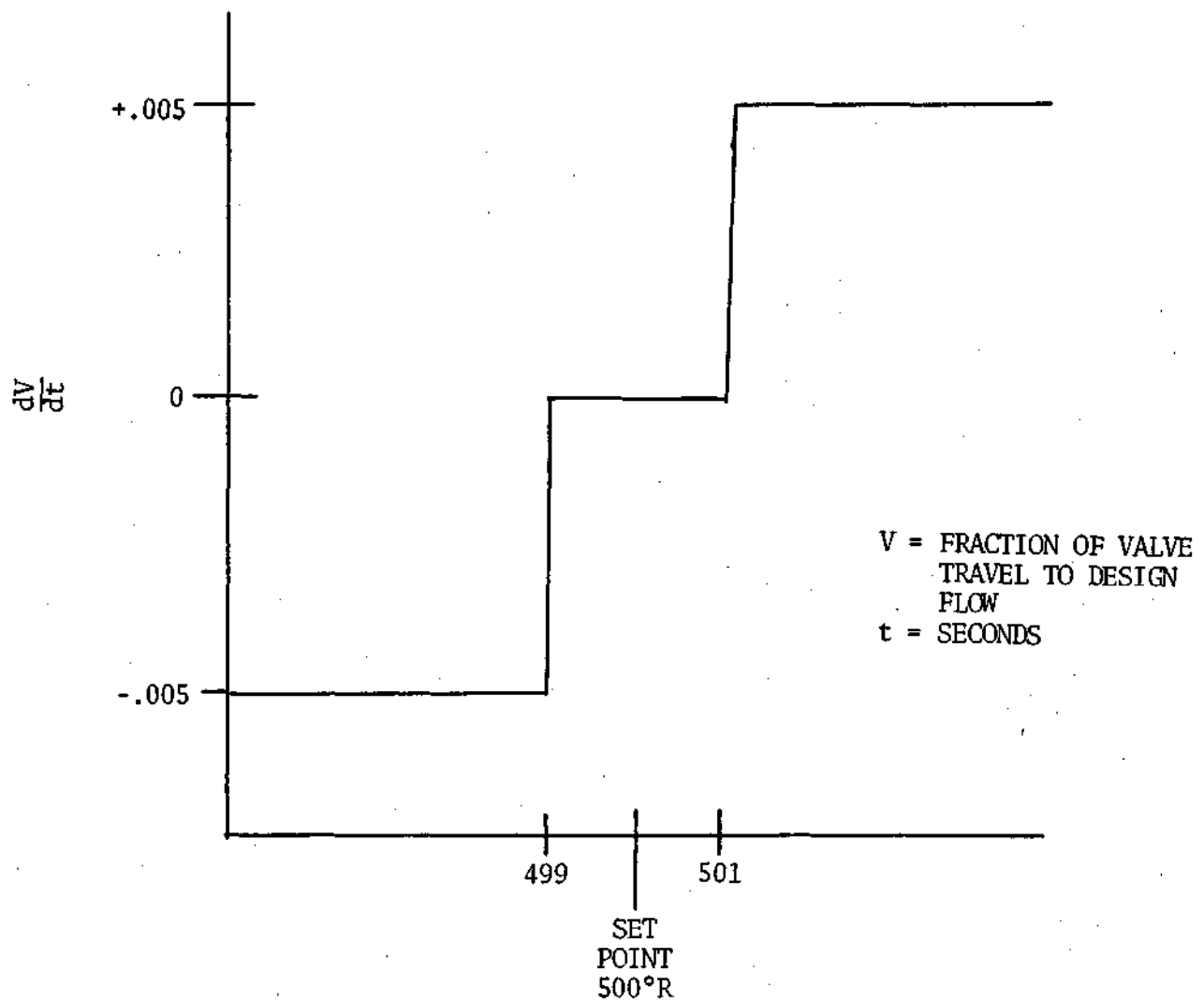
Non - linear flow versus valve stroke:

$$\frac{W}{W_{\text{design}}} = \frac{1}{.966} \left(\sin \frac{1.57}{1.2} \times \text{stroke} \right)$$

The non-linear H₂ flow versus valve stroke may arise due to valve design or from variations in pressure in the supply tank which changes flow versus stroke relationships by changing the density of H₂ passing through the valve. With a reasonably careful valve design, the non-linearities due to pressure changes should be within those assumed because the pressure in the relatively large supply tank cannot change at a high rate.

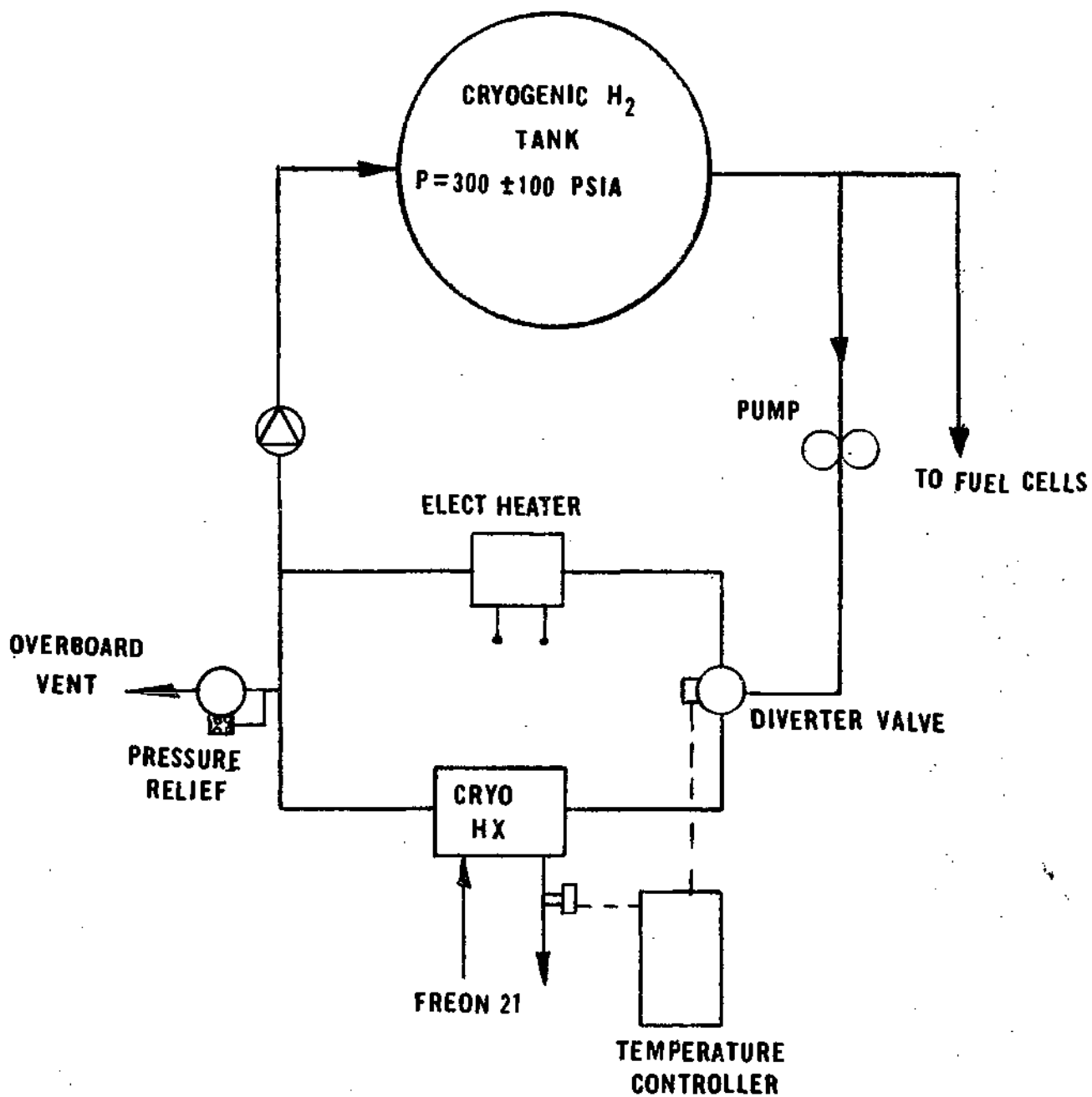
The type of control described was selected for its simplicity, since it requires only a constant speed reversible motor for its actuator; and for its low power consumption, since it requires no actuator power at steady state conditions. Using this type of control, the valve travel rate must be restricted to full travel in 200 seconds for stable operation given the tube - in - tube heat exchanger dynamic characteristics and the assumed non-ideal control characteristics. However, this valve travel rate was found to be adequate to control Freon outlet temperature with rates of change in Freon inlet temperature up to 20°R per minute. A 20°R per minute ramp was thought to be more severe than could occur in the Freon loop except during startup. Further study was recommended for the startup situation.

Under the present contract, the computerized model of cryogenic heat exchanger, controls, and Freon loop thermal dynamics described in the previous section was used as a tool to reevaluate the assumptions made in the



CRYOGENIC HEAT EXCHANGER CONTROL CHARACTERISTICS

FIGURE 9



CRYOGENIC SUPPLY SYSTEM SCHEMATIC

FIGURE 10

control selection, and to provide a definition of requirements during startup. Specific goals for this investigation were to:

- Confirm that during normal operation the rate of Freon temperature change at the cryogenic heat exchanger inlet does not exceed 20°R per minute for any appreciable time, and consequently that the selected control provides adequate response during normal operation.
- Determine whether Freon exit temperature perturbations caused by control action can return around the Freon loop in such a way as to be detrimental to control stability.
- Determine the effect of Freon exit temperature perturbations on the temperature of cooling water entering the cabin condensing heat exchanger and Freon entering the fuel cell heat exchangers. This determination allows a more realistic specification of control limits, especially during startup.
- Define the startup transient and modify the control as required to provide acceptable temperature control limits during the startup.

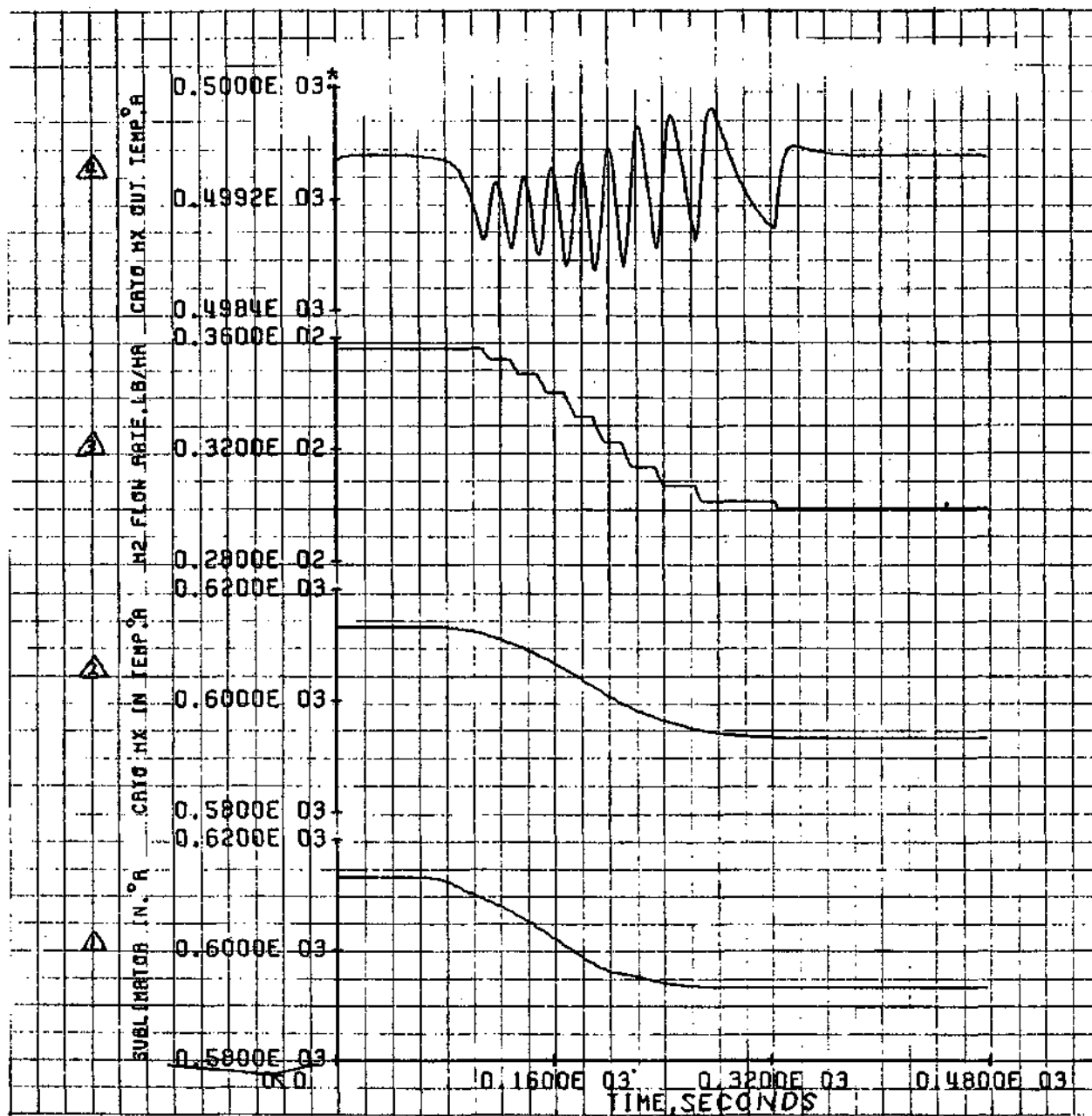
Two input disturbances were applied to evaluate the action of the cryogenic heat exchanger control under non-startup conditions: 1) starting with design heat loads, one fuel cell was "shutdown" by lowering the temperature of FC 43⁽¹⁾ entering one fuel cell heat exchanger in a two minute period; and 2) again starting with design heat loads, one fuel cell and the cabin water loop were simultaneously "shutdown" in a two minute period. The second disturbance is more severe than can actually occur because the thermal mass of the water loop will prevent cooling of the water at this high rate.

The startup transient was investigated by setting design heat loads on the interface and fuel cell heat exchangers, with the Freon in the loop from the beginning of the radiator bypass through the inlet to the interface heat exchanger at 500°R. This is the situation at the instant the radiator bypass valve is opened.

The results of these investigations are presented in computerized plots in figures 11 through 14. The location of each variable plotted is located by number on the schematic in figure 5. Some general remarks about certain features of the thermal dynamics of the system may help in interpreting these plots:

- Transport time of the liquid around the loop is on the order of 150 seconds. Heat added to a particle of Freon cannot result in a temperature rise in a downstream component until after the period of time it takes the fluid particle to travel to that component.

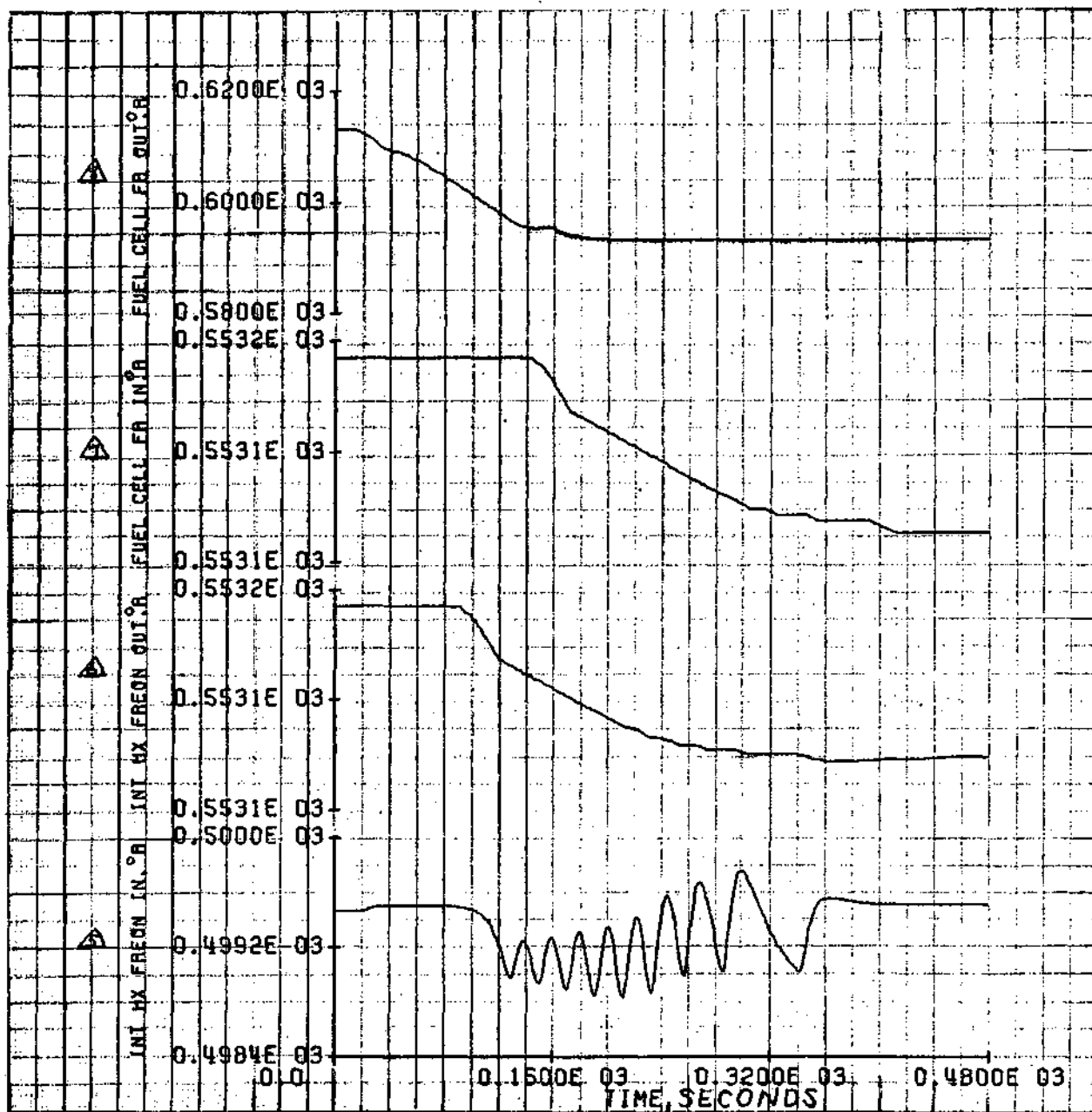
⁽¹⁾ FC 43 is a 3M Company coolant.



* Signifies exponent i.e., 10^3 .

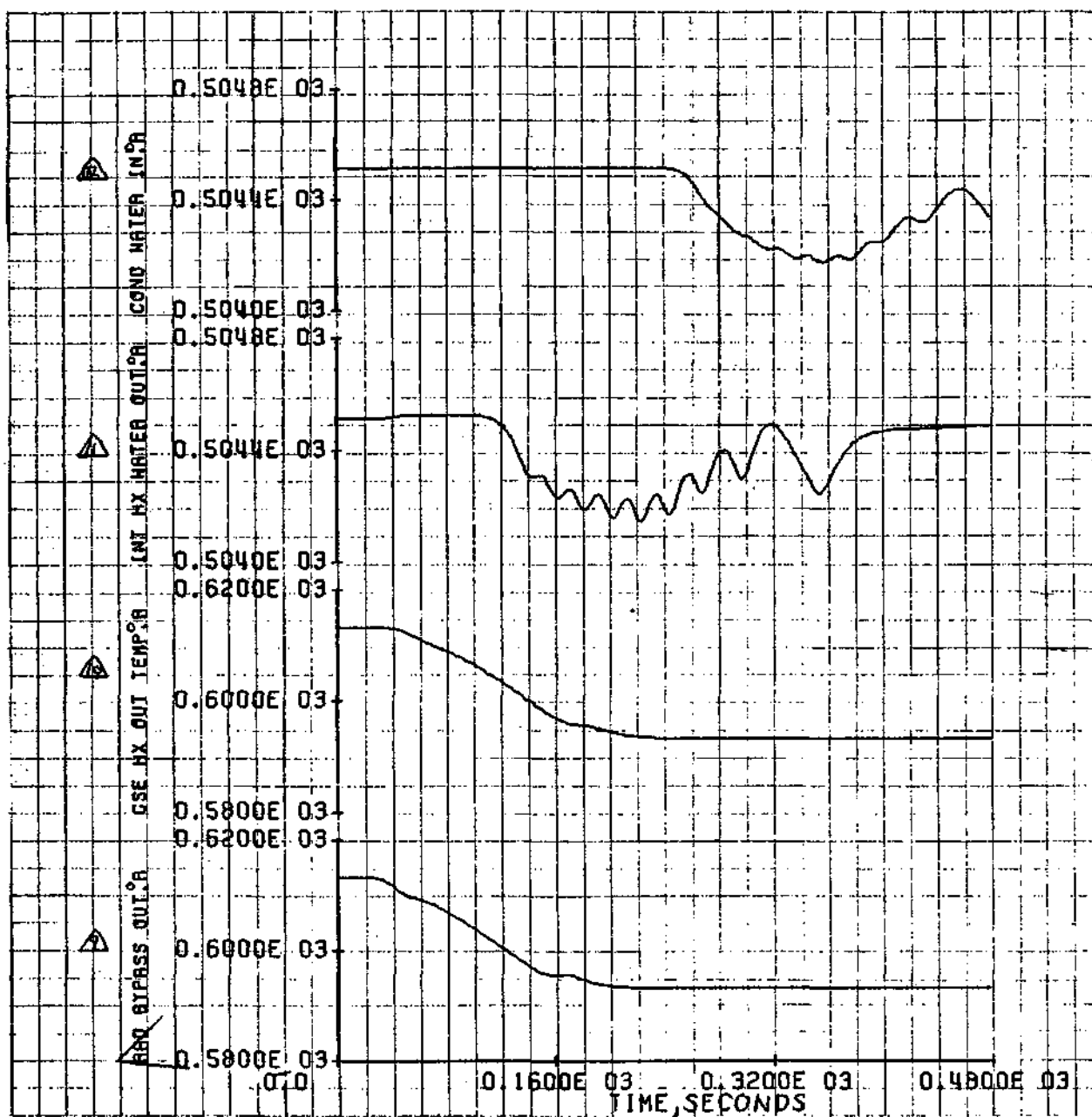
TRANSIENT RESPONSE - SHUTDOWN ONE FUEL CELL

FIGURE 11-A



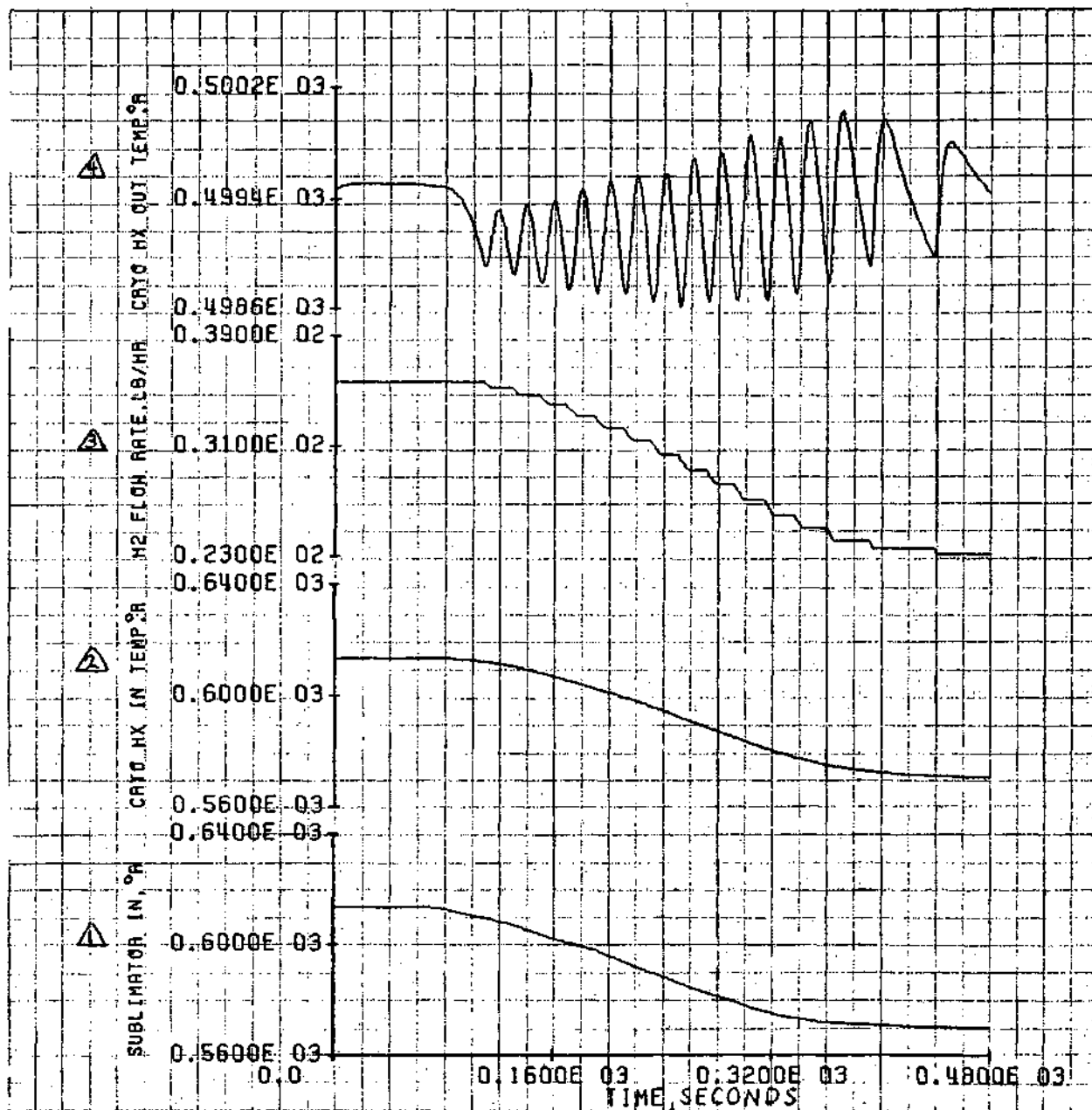
TRANSIENT RESPONSE - SHUTDOWN ONE FUEL CELL

FIGURE 11-B



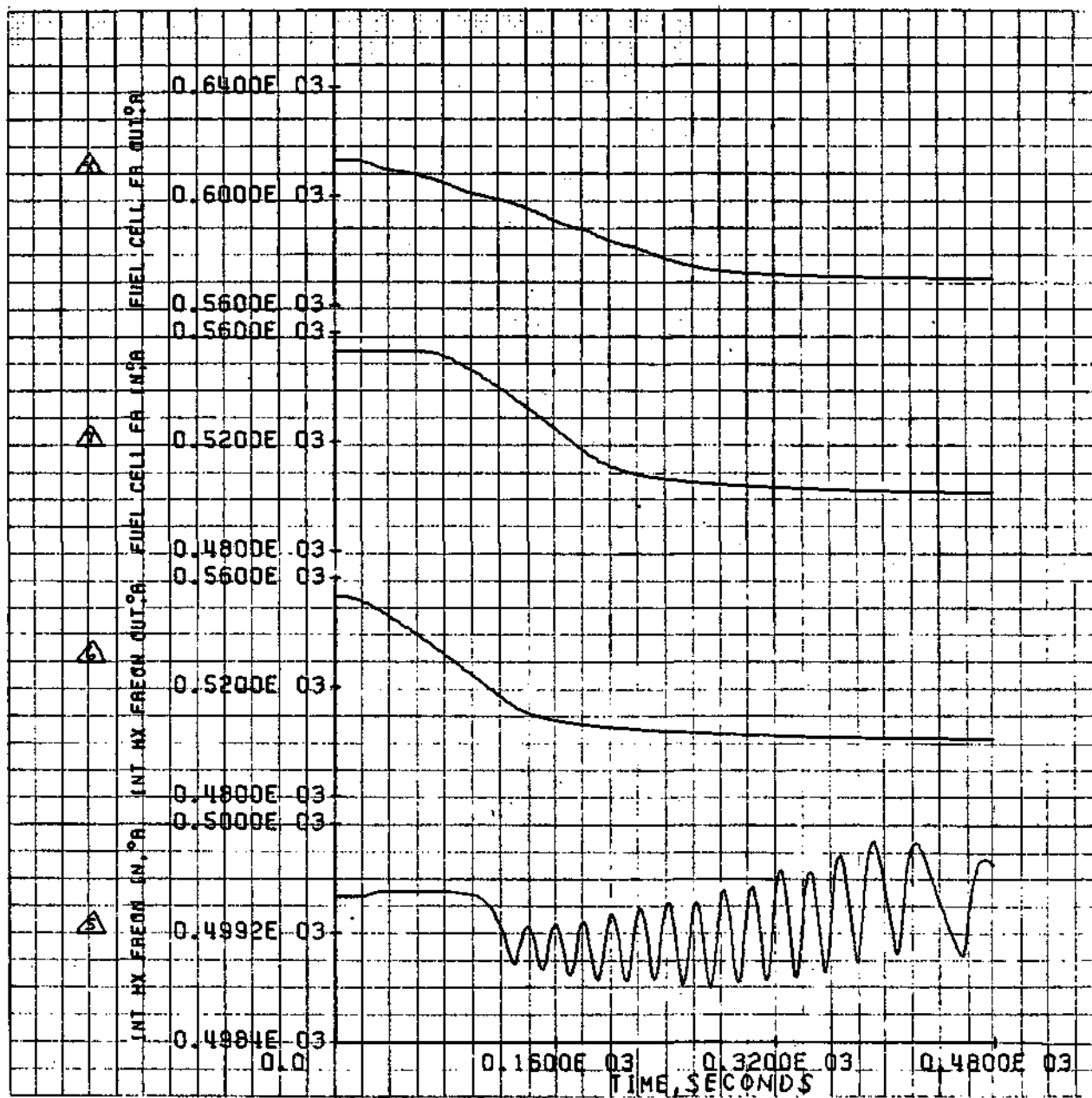
TRANSIENT RESPONSE - SHUTDOWN ONE FUEL CELL

FIGURE 11-C



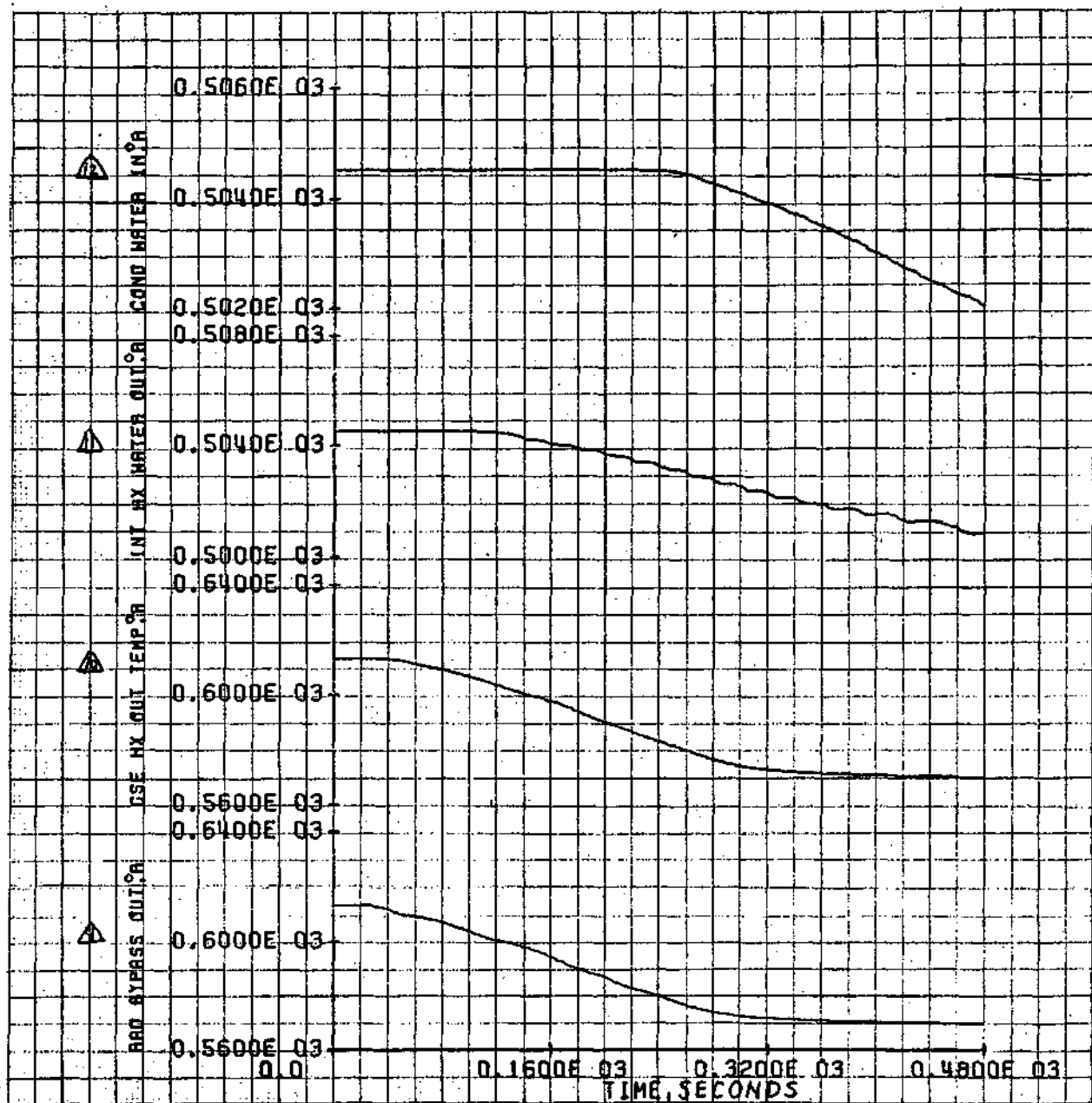
TRANSIENT RESPONSE - SHUTDOWN FUEL CELL & H₂O LOOP

FIGURE 12-A



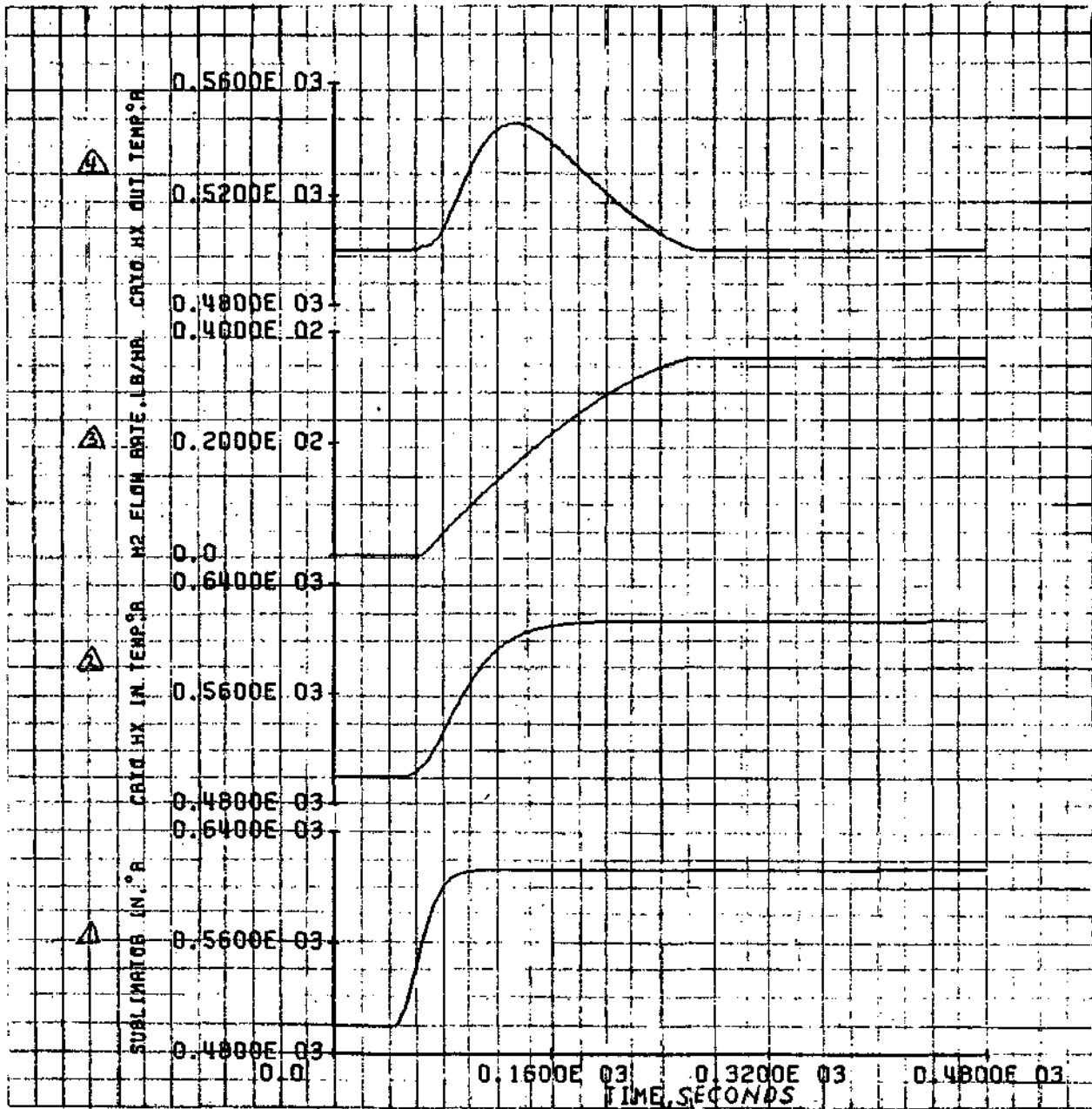
TRANSIENT RESPONSE - SHUTDOWN FUEL CELL & H₂O LOOP

FIGURE 12-B



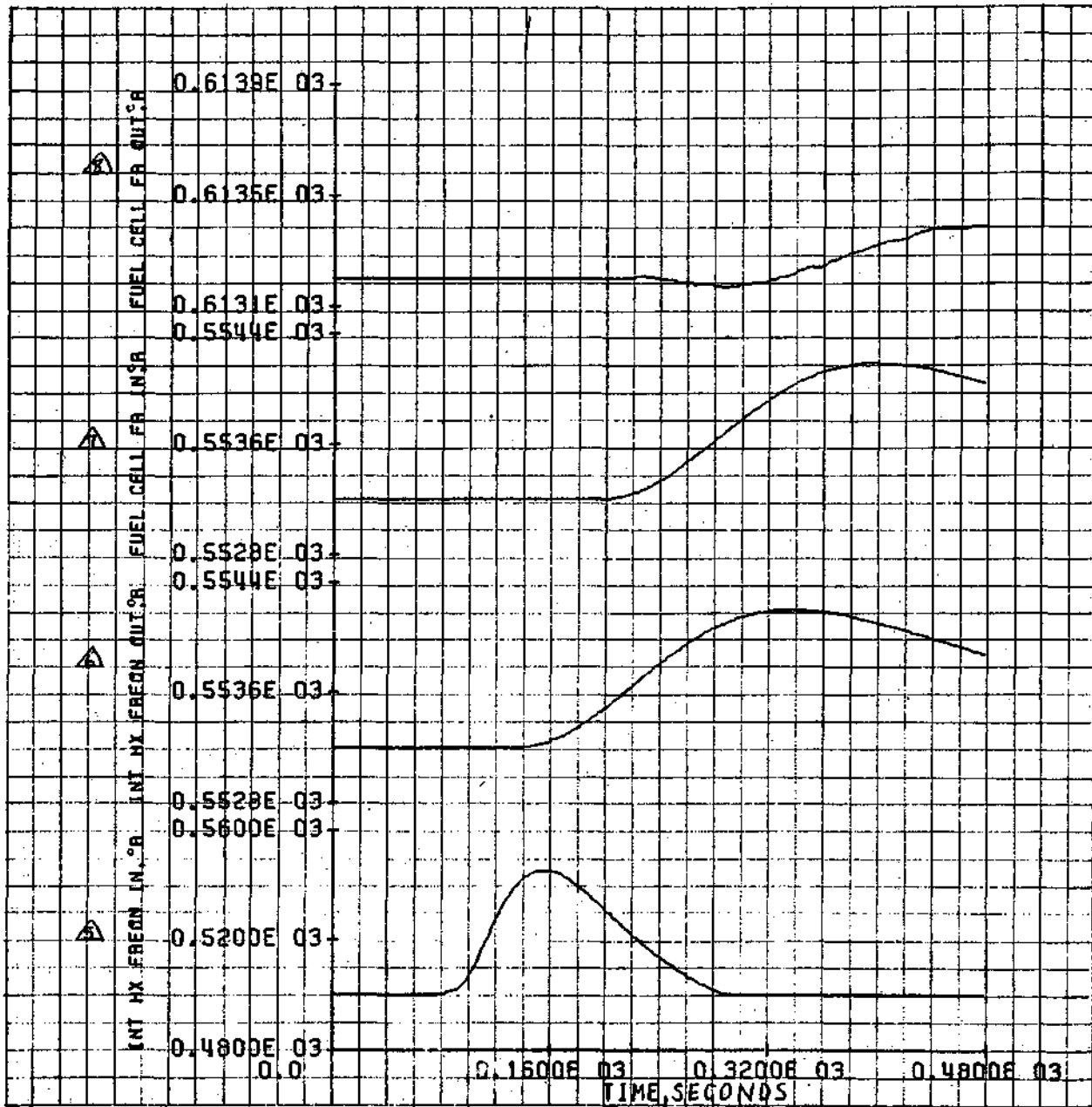
TRANSIENT RESPONSE - SHUTDOWN FUEL CELL & H₂O LOOP

FIGURE 12-C



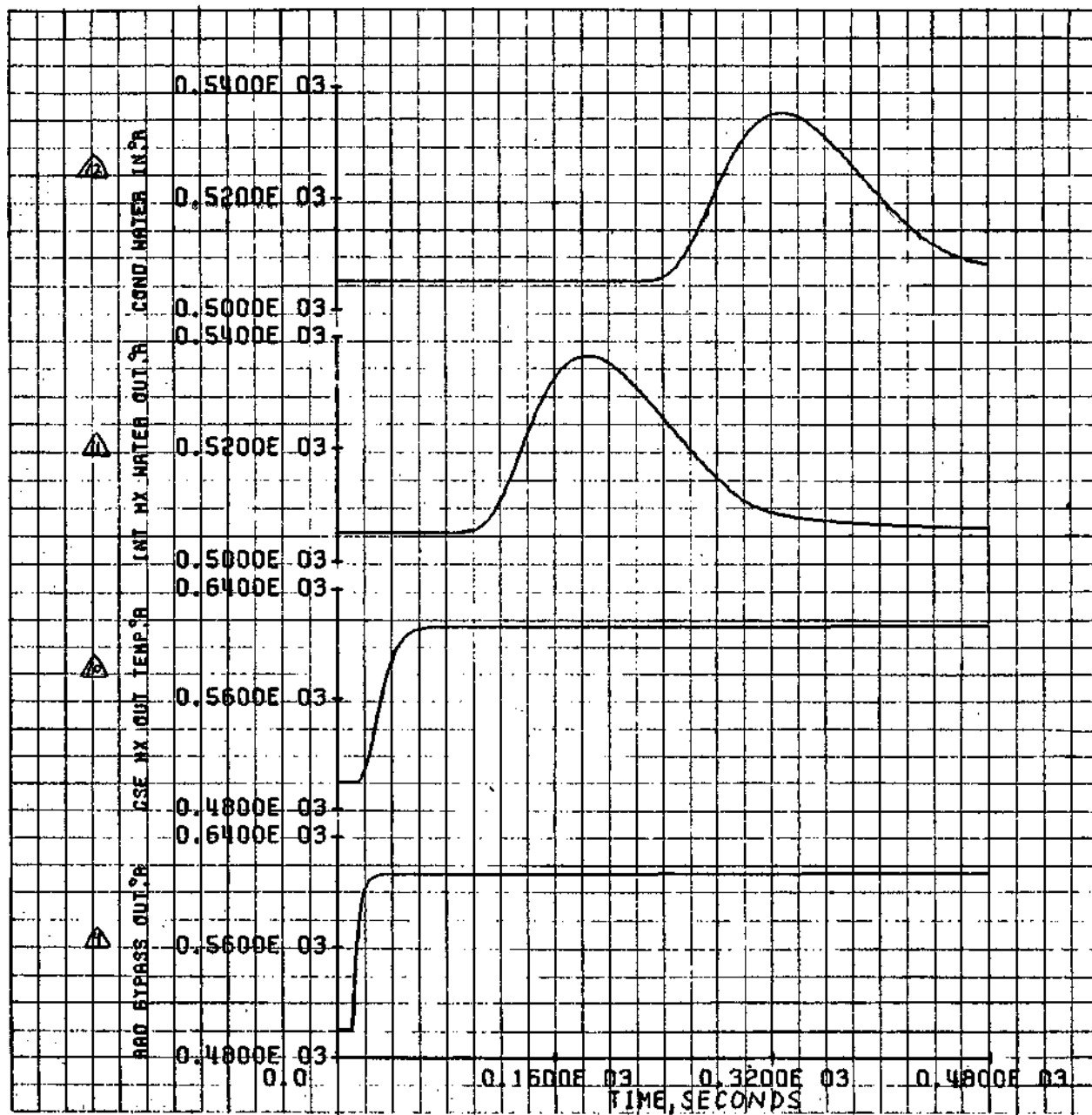
TRANSIENT RESPONSE - STARTUP WITHOUT LEAD

FIGURE 13-A



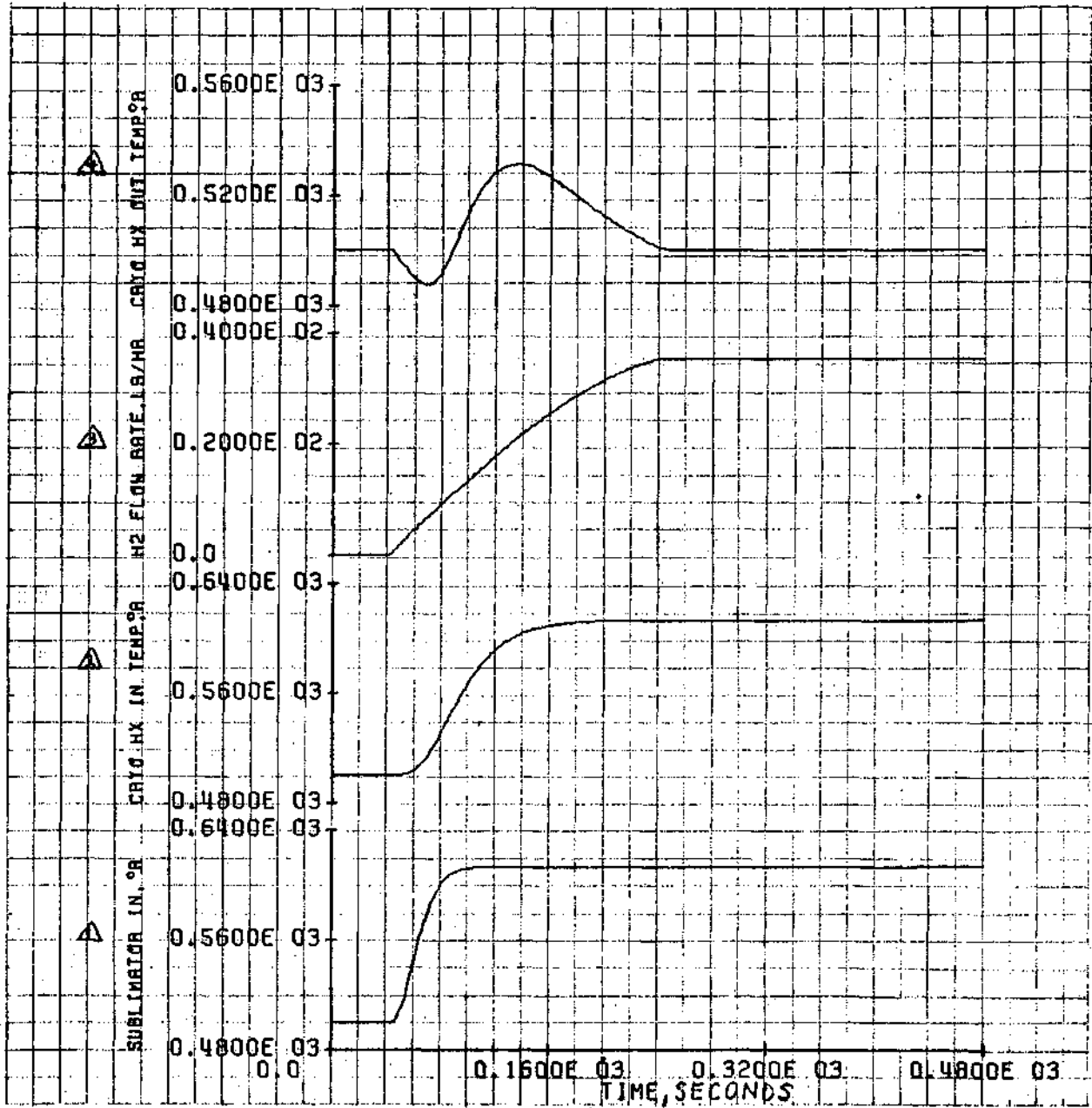
TRANSIENT RESPONSE - STARTUP WITHOUT LEAD

FIGURE 13-B



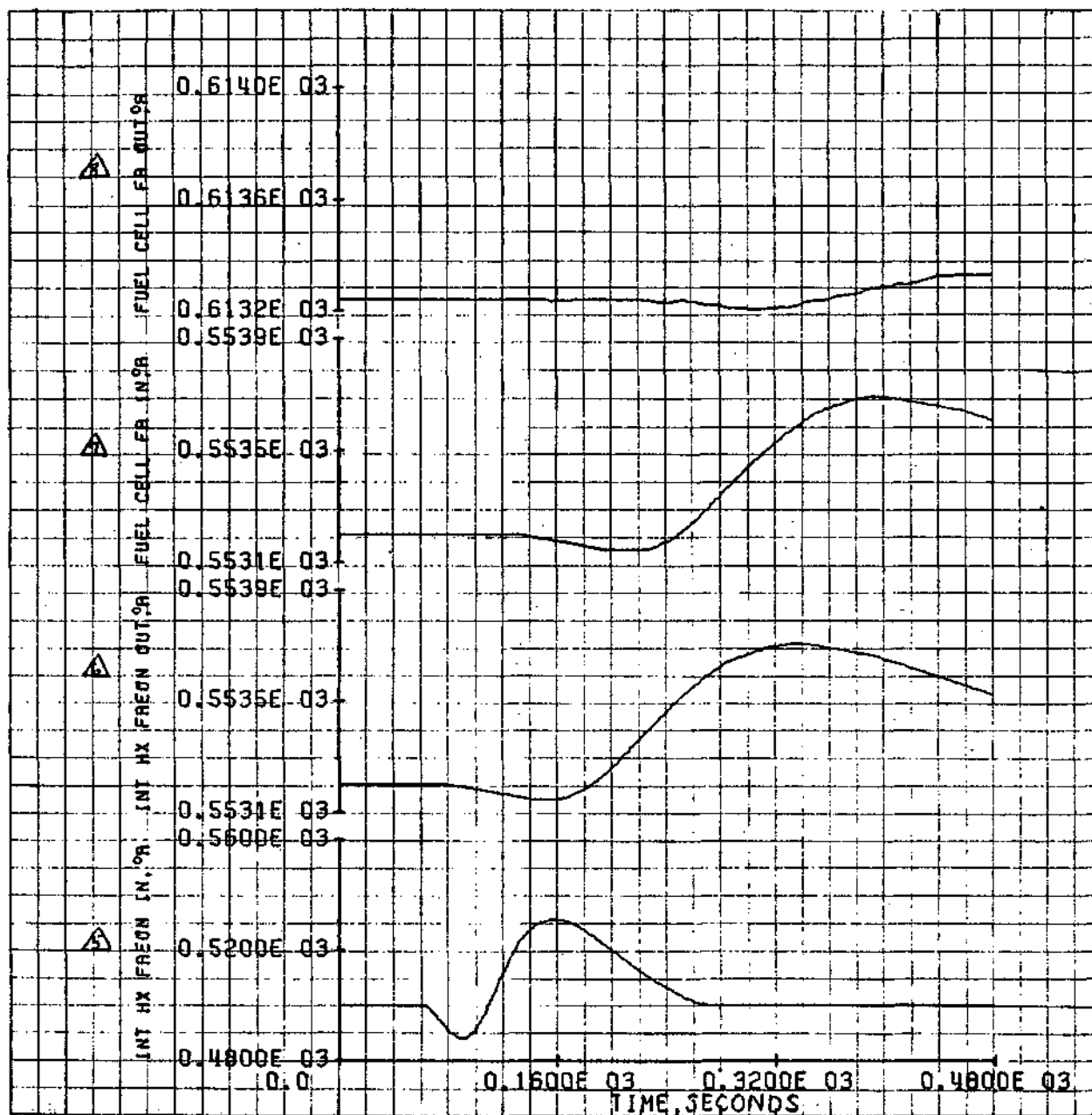
TRANSIENT RESPONSE - STARTUP WITHOUT LEAD

FIGURE 13-C



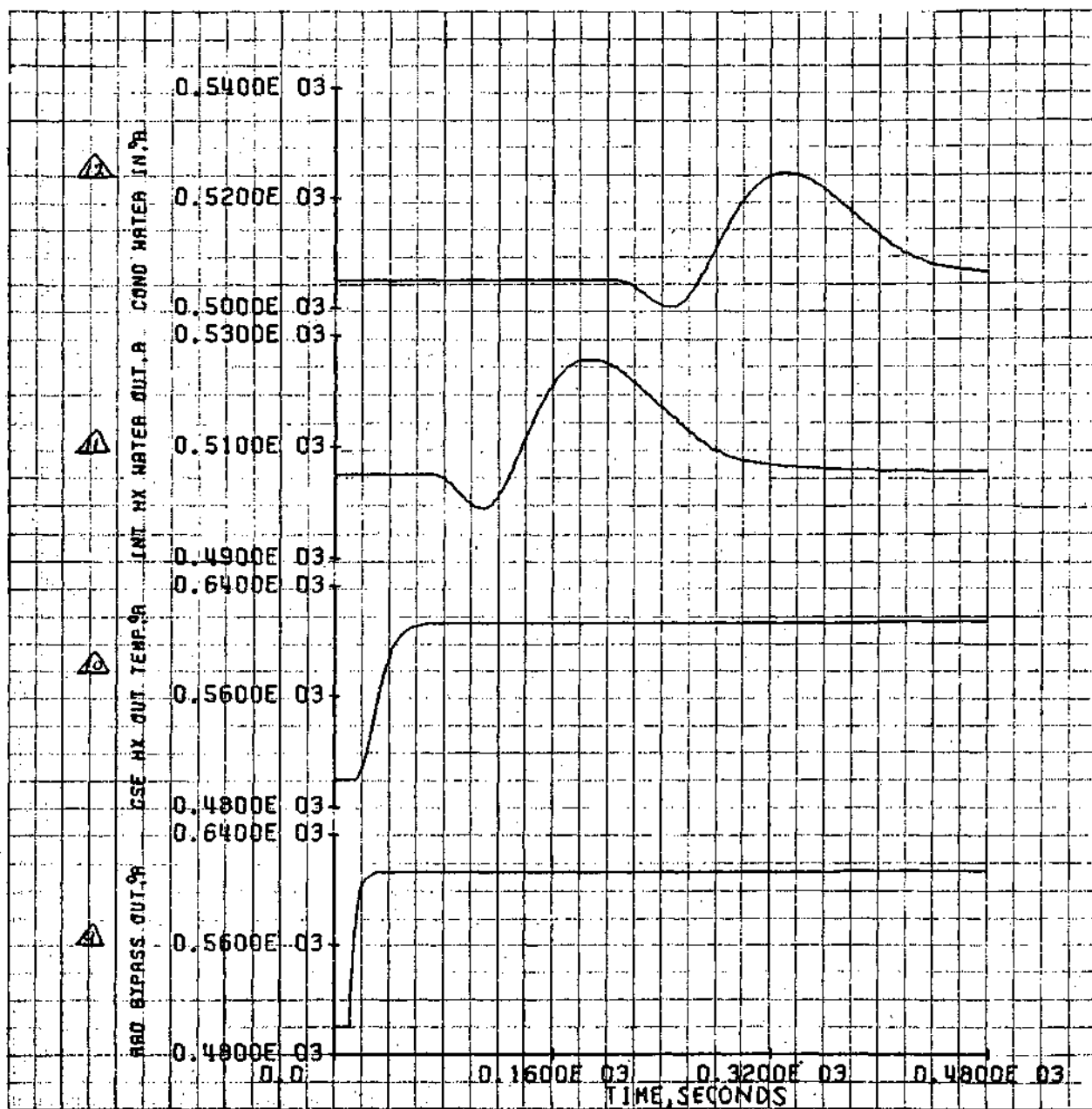
TRANSIENT RESPONSE - STARTUP - MODERATE LEAD

FIGURE 14-A



TRANSMITH RESPONSE - STARTUP - MODERATE LEAD

FIGURE 14-B



TRANSIENT RESPONSE - STARTUP - MODERATE LEAD

FIGURE 14-C

- The fuel cell heat exchangers and especially the interface heat exchanger are highly effective. The steady state change in the temperature leaving a heat exchanger is one minus the effectiveness times the change in incoming temperature of that fluid (assuming no temperature change in the other fluid). This value is .085 for the interface heat exchanger.
- The sum of thermal mass in the Freon loop is large, and the sum of metal time constants around the loop is of the same order of magnitude as the control valve travel time.

Each of the plots is discussed below.

Shutdown of One Fuel Cell (Figure 11)

Starting with the design heat load on the cryogenic heat exchanger, the shutdown of one fuel cell was simulated by linearly reducing the temperature of FC 43 coolant entering one fuel cell heat exchanger from 629.2°R to 553.2°R over a period of two minutes beginning at time zero. The resulting temperature transient and control action are shown in figure 11.

The maximum rate of change of Freon temperature at the cryogenic heat exchanger inlet is about 8°R per minute, well under the 20°R per minute the control was designed to handle. Consequently the Freon outlet temperature does not go outside the 499°R to 501°R control dead band by more than 0.3°R, well within tolerances of $\pm 5^\circ\text{R}$.

A comparison of Freon temperatures entering and leaving the interface heat exchanger (plots 8 and 9) shows that none of the "high frequency" temperature fluctuations caused by control action are able to pass through the large, high effectiveness interface heat exchanger. Therefore, there is no possibility of unstable control action resulting from temperature feedback around the Freon loop.

Simultaneous Shutdown of One Fuel Cell and Cabin Water Loop (Figure 12)

Figure 12 shows the results of reducing the inlet FC 43 temperature of one fuel cell from 629.2°R to 553.2°R and reducing the water temperature into the interface heat exchanger from 558.2°R to 500°R over two minutes starting at time zero. Although the transient lasts much longer than when a single fuel cell is shutdown, the rate of change in Freon temperature at the cryogenic heat exchanger inlet is scarcely greater. This is the result of the transport delay between the time a particle of Freon enters the interface heat exchanger and the time the same particle enters the fuel cell heat exchanger.

Only the initial portion of the plotted transient (between 300 and 400 seconds) is a realistic representation of the response to the input described above. This is so because the fuel cell cooling loop has not been included in the computer model and the FC 43 temperature at the inlet to the active fuel cell heat exchangers was constant. Without adjustment in FC 43 loop temperatures to changes in Freon temperature, the correct steady state is not approached at the end of the transient.

The input disturbance in this case is more severe than can be imagined to occur in reality, because the large thermal mass in the cabin water loop would prevent water temperature reduction at the rate input. Consequently the control described at the beginning of this section is more than adequate in non-startup situations.

Startup (Figures 13 and 14)

The most severe Freon temperature changes at the cryogenic heat exchanger inlet will occur when the radiator is effectively removed from the Freon loop prior to descent, and the full heat rejection load is imposed on the heretofore inactive cryogenic heat exchanger.

The cryogenic heat exchanger H_2 control valve will initially be closed and will require 200 seconds to get to a full-open position.

The radiator may be removed from the system by immediately bypassing Freon around the radiator, or by first stowing the radiator and bypassing it at some later time. If the radiator is stowed before bypassing, the heat load will be imposed on the cryogenic heat exchanger more gradually because the large radiator metal mass will provide cooling for a period of time after heat rejection by radiation to space has stopped. At the present time, there appears to be no disadvantage to first stowing the radiator. However, since bringing the cryogenic heat exchanger on-line by suddenly bypassing the radiator is the most severe startup mode, it is the one considered in this study.

The startup transient that results with use of the control previously described is shown in figure 13. The Freon temperature at the cryogenic heat exchanger exit rises to $545^\circ R$, or 45° above the control temperature. It is outside the control dead band for a period of about 200 seconds. The effect of this startup temperature excursion on the operation of the fuel cells and the cabin condensing heat exchanger may be estimated from figure 13. Figure 13 shows the Freon temperature at the inlet to the fuel cell heat exchangers rises less than a degree, and therefore, startup will have negligible effect on the fuel cells. The small temperature rise at the fuel cell heat exchanger's inlet occurs because the interface heat exchanger acts as a buffer.

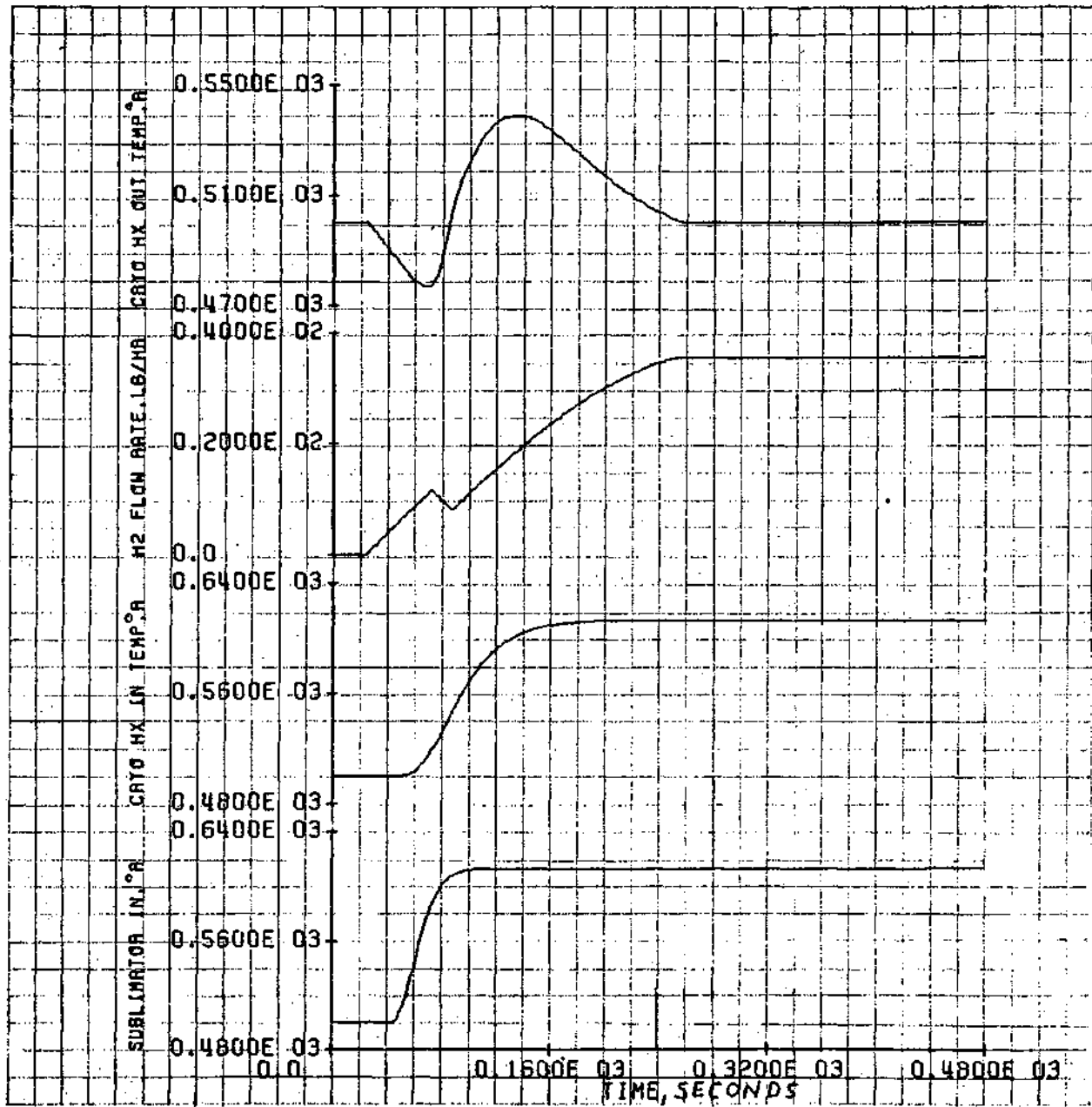
The cooling water entering the cabin condensing heat exchanger, shown in figure 15, rises 30°R above its design value to 535°R. This will cause a temporary rise in cabin humidity because some of the water which has been condensed in the heat exchanger will be re-evaporated. The duration of high cooling water temperature is so short, however, that it is unlikely that serious discomfort will result.

Although the startup transient shown in figure 13 may be acceptable, the magnitude of temperature excursions during startup can be reduced by the addition of simple anticipation or lead to the controller. By obtaining the time derivative of Freon temperature at some point upstream of the cryogenic heat exchanger, appropriate control action can be taken before a temperature change is sensed at the cryogenic heat exchanger Freon loop. As an example, two thermocouples could be placed a measured distance apart in line 4 (see figure 5). The difference between the two thermocouple signals would be proportional to rate of change in Freon temperature which will appear at the cryogenic heat exchanger inlet at some later time. When the rate of change exceeds a certain value, the controller causes the actuator to begin opening the H₂ control valve. This anticipation is required only during startup so the rate of change that activates the control would be set at 20°R per minute.

Figure 14 shows the startup transient using such anticipation in the controller. The temperature rise in cooling water at the condenser inlet is reduced to 19 degrees, with a maximum temperature of 524°R occurring. The locations of the anticipation thermocouples in this case were about 5 and 5-1/2 feet upstream of the sublimator. These locations appear reasonably close to optimum since an attempt to increase the amount of lead by moving the thermocouples 29 feet upstream produced the control action plotted in figure 15. The valve started to open earlier (as indicated by the H₂ flow rate), but the temperature derivative went below its threshold value before the temperature at the cryogenic heat exchanger exit rose to the lower limit of the control band. As a result, the valve started to close again, the exit temperature rose sharply, and the resulting temperature excursion is of the same magnitude as in the previous case.

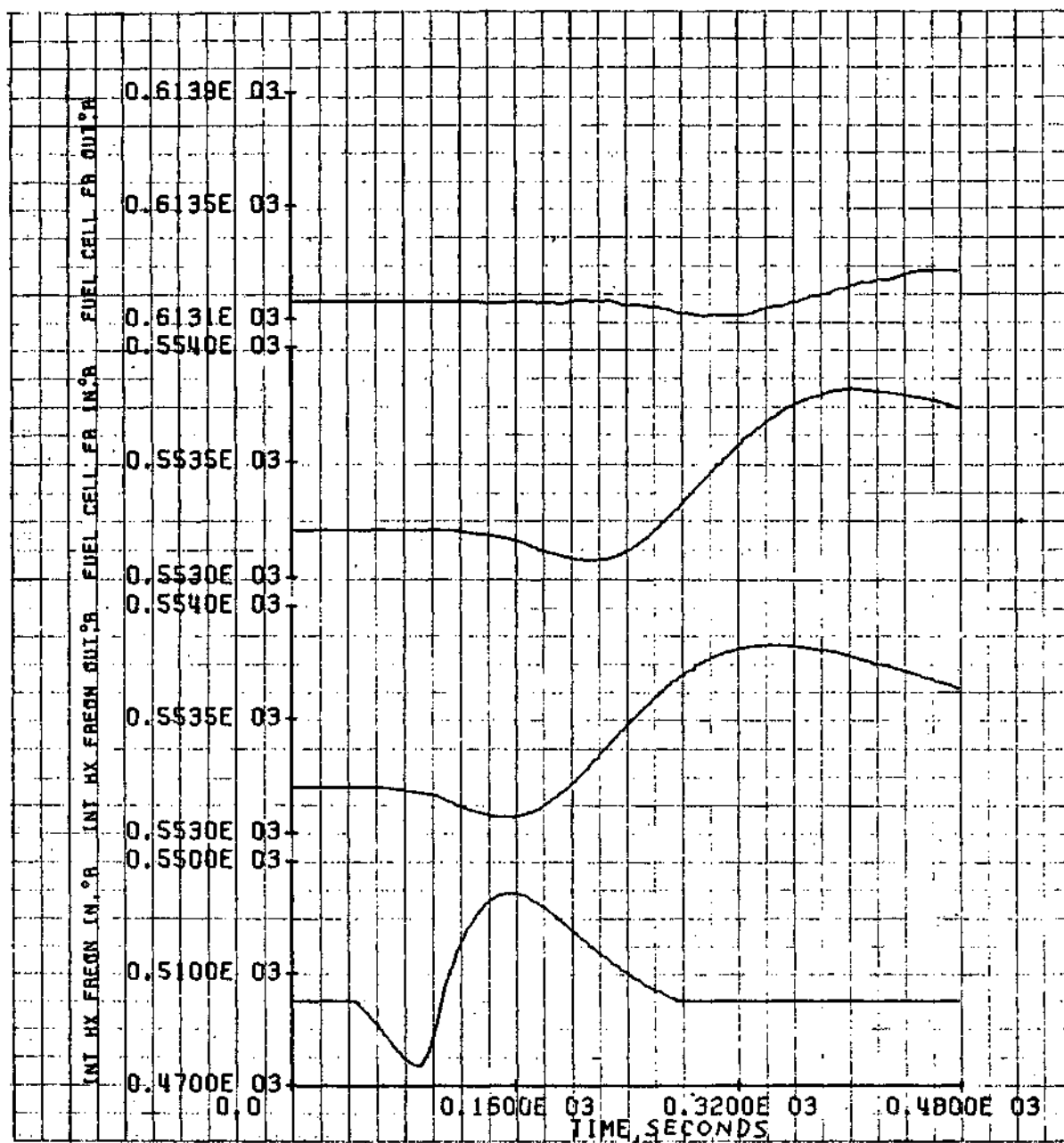
Conclusions of Study

- A constant speed control valve actuator with a rate of full travel in 200 seconds provides adequate temperature control in all non-startup situations.
- During the most severe startup mode such a control allows a 30°R temperature rise in cooling water entering the cabin condensing heat exchanger. The temperature is above its design value for a period of 200 seconds. This short time excursion seems acceptable, but its magnitude may be reduced to 19°R or less by adding anticipation to the controller.



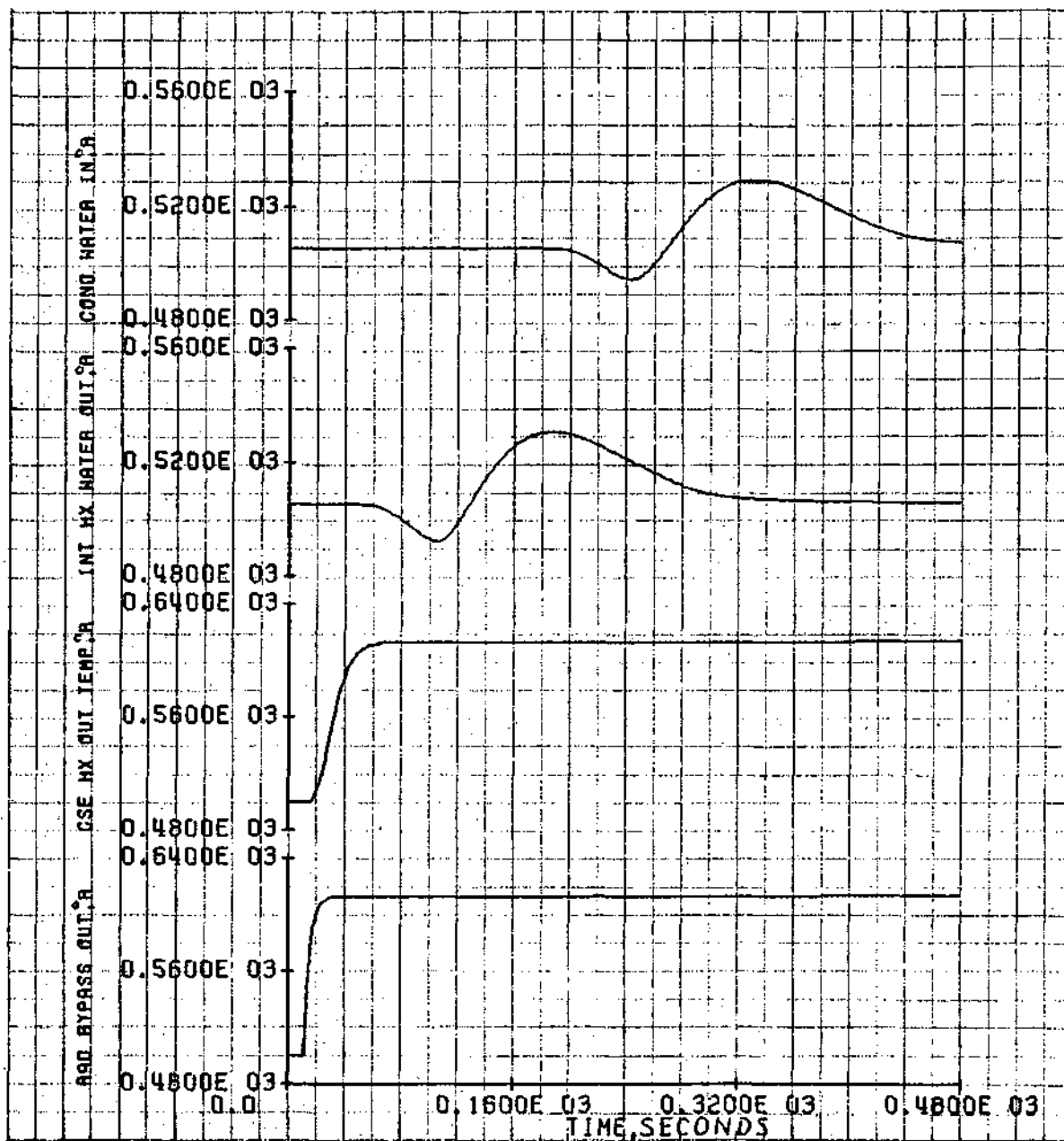
TRANSIENT RESPONSE - STARTUP - MAXIMUM LEAD

FIGURE 15-A



TRANSIENT RESPONSE - STARTUP - MAXIMUM LEAD

FIGURE 15-B



TRANSIENT RESPONSE - STARTUP - MAXIMUM LEAD

FIGURE 15-C

- No temperature fluctuations caused by control action can feed back through the Freon loop.

Heat Exchanger Breadboard Design

Using information and stress analysis conducted under the design study, the heat exchanger was depicted in drawing SVSK 86021. This design was evolved after discussions with Jones Metal Products Company, who fabricated the final heat exchanger. The design was reviewed with the NASA on August 23, 1972 and approval to manufacture was obtained.

Heat Exchanger Procurement

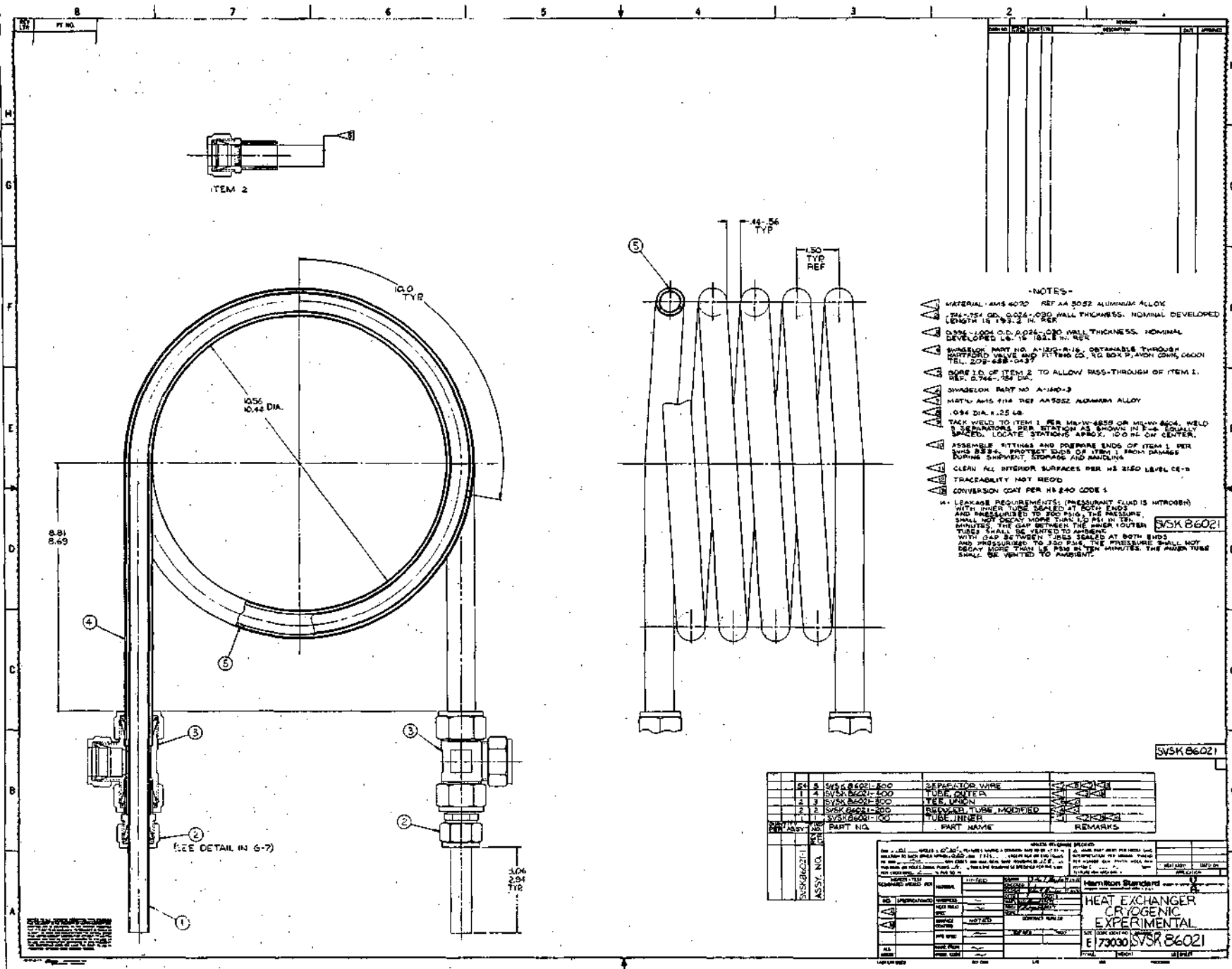
Requests for bid on the tube - in - tube cryogenic heat exchanger depicted on SVSK 86021 were sent to three outside vendors and to the Hamilton Standard manufacturing facility. Only one affirmative reply was received, from the Jones Metal Products Company of East Windsor, Connecticut. The remainder either did not have the specific tube rolling capability or did not have the current shop capacity to handle this job. Therefore, a purchase order was placed with Jones Metal Products Company.

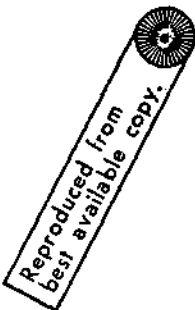
Unfortunately twelve feet was the maximum tubing length available without purchasing a mill run. The first attempt at manufacture was to butt weld two lengths of tubing. The initial attempt to manufacture the heat exchanger disclosed two problems:

1. Fracture occurred during coiling at the butt weld used to extend the tubing length.
2. Severe rippling of the outer tube occurred at the inside of the coil.

In order to eliminate the weld fracture problem the heat exchanger length was reduced as shown in SVSK 86021, Revision A. As can be seen in figure 2, this results in a configuration with a system weight approximately one pound more than optimum but allows the basic objective of the program, which is correlation of thermal and dynamic analysis with test. Should prototype implementation be required, time and funds should be allotted to obtain a mill run of the proper length.

To correct the rippling problem the thickness of the outer tube was increased from .028 inches to .035 inches. This resulted in a marginally acceptable condition where minor dents in the tubing served as the focal point for rippling. Fortunately with careful rolling and working of the tubing rippling was held to an acceptable limit. Future designs utilizing this tubing should have the coil diameter increased from 10.5 inches to about 12.0 inches.





The heat exchanger was received at Hamilton Standard and subjected to 300 psig leakage/pressure tests and was found acceptable. The completed heat exchanger is depicted in figure 16 and the x-ray shown in figure 17 shows the separation between inner and outer tubes. It was then cleaned to HS 3150 CE-3 in the hydrogen loop and HS 2241 G in the Freon loop.

Set-Up Test Facility

Upon completion of the system dynamic analysis, meetings were held with design, analysis and facilities engineering to define the detailed test requirements and requisite facilities. The test requirements are contained in Appendix A. It should be noted that any valves required to perform transients were considered as part of the test equipment and no simulation of the closed loop temperature control valve, controller, or sensor would occur within this test program. The transients defined in Appendix A are acceptable to evaluate dynamic response and obtain a thermal transfer function.

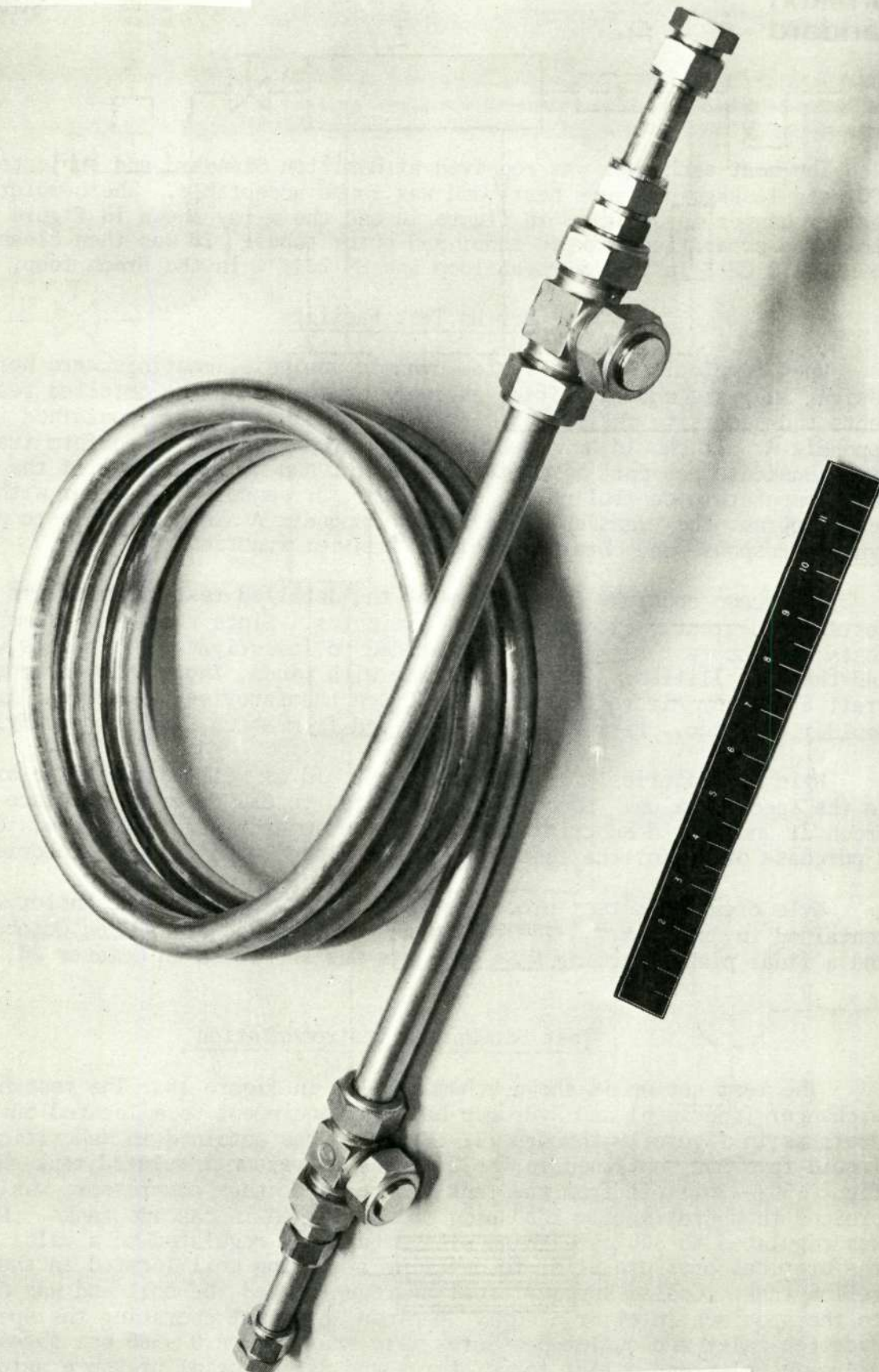
It became apparent from review of the detailed test requirements that test costs would greatly exceed original estimates. Since the major element of these costs was the test set-up, it was decided to investigate test houses who already had these facilities. Contact was made with Linde, Inc., Wyle Laboratories, Pratt & Whitney Aircraft of Florida, Ogden Laboratories, and Beech Aircraft of Boulder, Colorado. Firm bids were received from Beech Aircraft and Wyle Laboratories.

Wyle Laboratories provided the lowest bid as well as taking no exception to the specification. Beech Aircraft wanted to use Freon 11 in place of Freon 21 and wanted subcritical rather than supercritical hydrogen. Consequently, a purchase order for the test program was placed with Wyle Laboratories.

Wyle prepared a test procedure which is included in the Master Test Plan contained in Appendix B. The test plan was reviewed by NASA on October 11, 1972 and a final plan including NASA comments was submitted on October 24, 1972.

Test Set-Up and Instrumentation

The test set-up is shown schematically in figure 18. The test heat exchanger (specimen) and hydrogen handling equipment were located out-of-doors as in figures 19 through 22. Hydrogen was obtained in subcritical liquid form and contained in the 2000 gallon vacuum insulated tank shown in figure 20. Boil off from the tank entered a 5 stage compressor, was compressed to approximately 2300 psig and contained in cascade tanks. Pressure was regulated to 500 psig in an ullage tank and regulated to a value above the critical pressure prior to entering a cooling coil located in the 2000 gallon tank. Cooled supercritical hydrogen exited the coil and was directed to the specimen inlet or a dummy specimen while not operating the specimen. Hydrogen inlet and outlet pressures were measured on 0-300 psi gages. Hydrogen inlet and outlet temperatures and differential pressure across the specimen were measured and recorded on Mosely recorders shown in figure 23.



HYDROGEN - FREON 21 CRYOGENIC HEAT EXCHANGER

FIGURE 16

Hamilton
Standard

U
A
DIVISION OF UNITED AIRCRAFT CORP.

SVHSR 6180

B

C 2 3 A B
HAMILTON STANDARD DIV
WINDSOR LOCKS CONN
1 0 2 6 7 2 P

TV

12

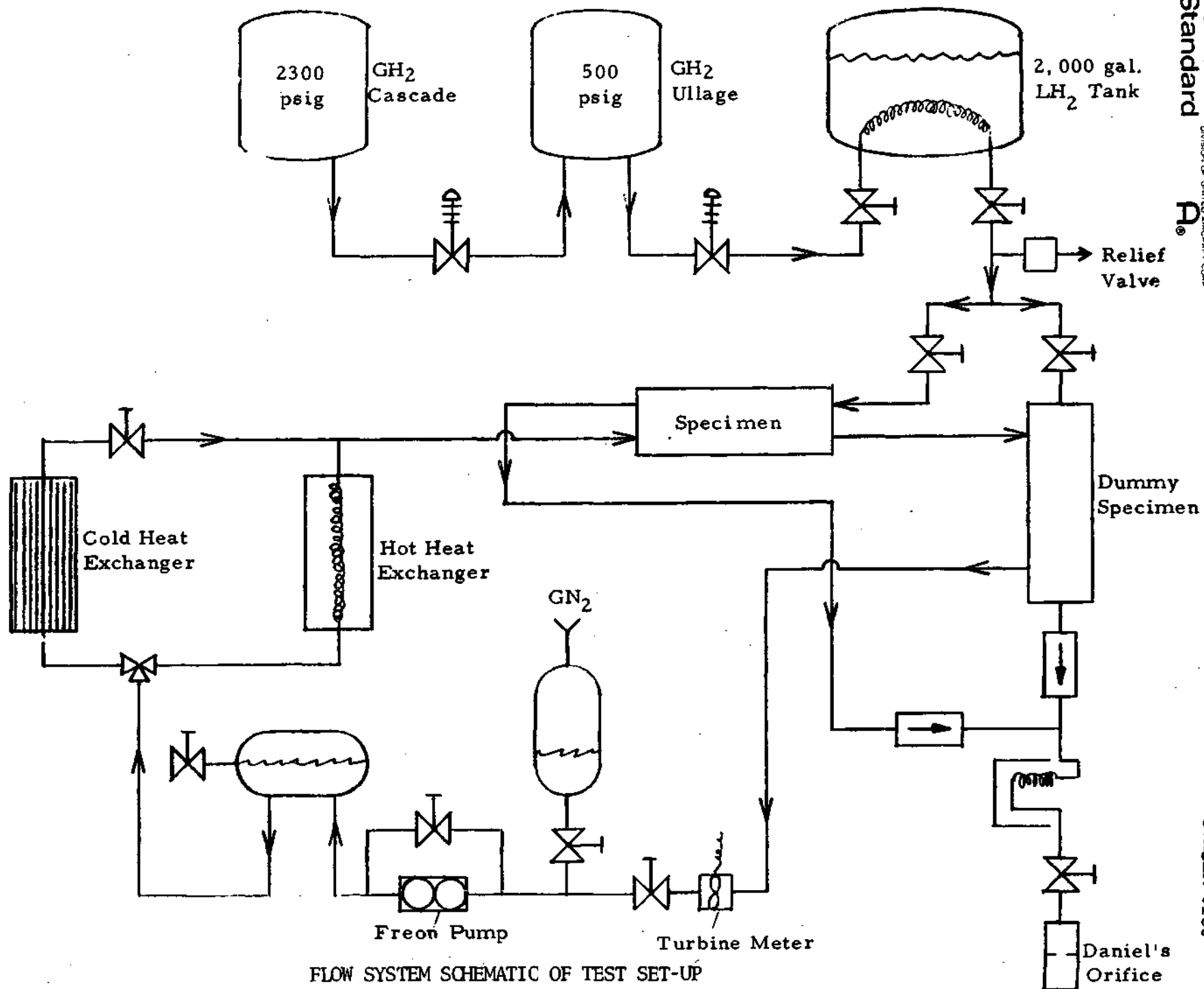
A

X-RAY OF HYDROGEN - FREON 21 CRYOGENIC HEAT EXCHANGER

FIGURE 17

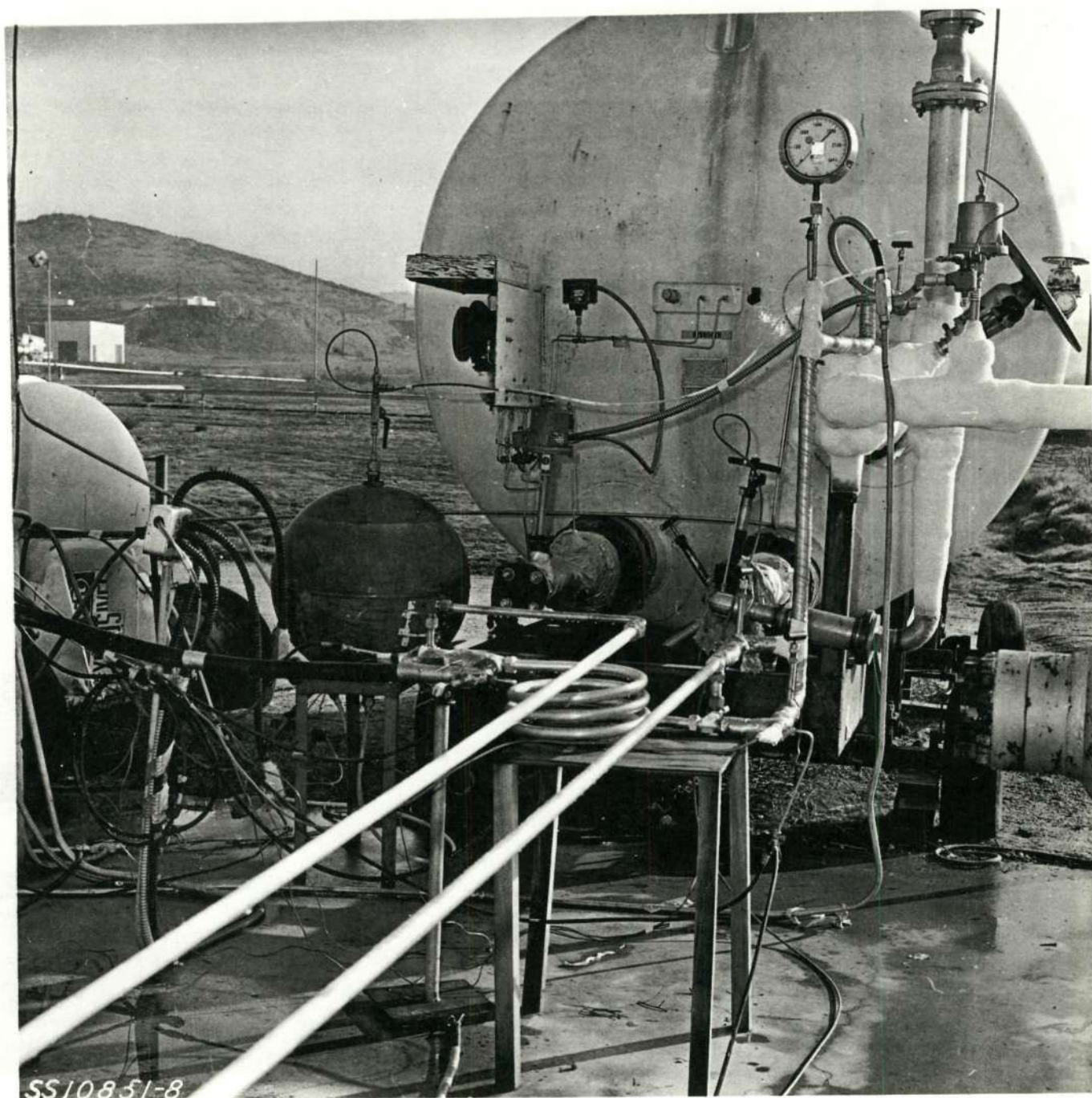
65

F



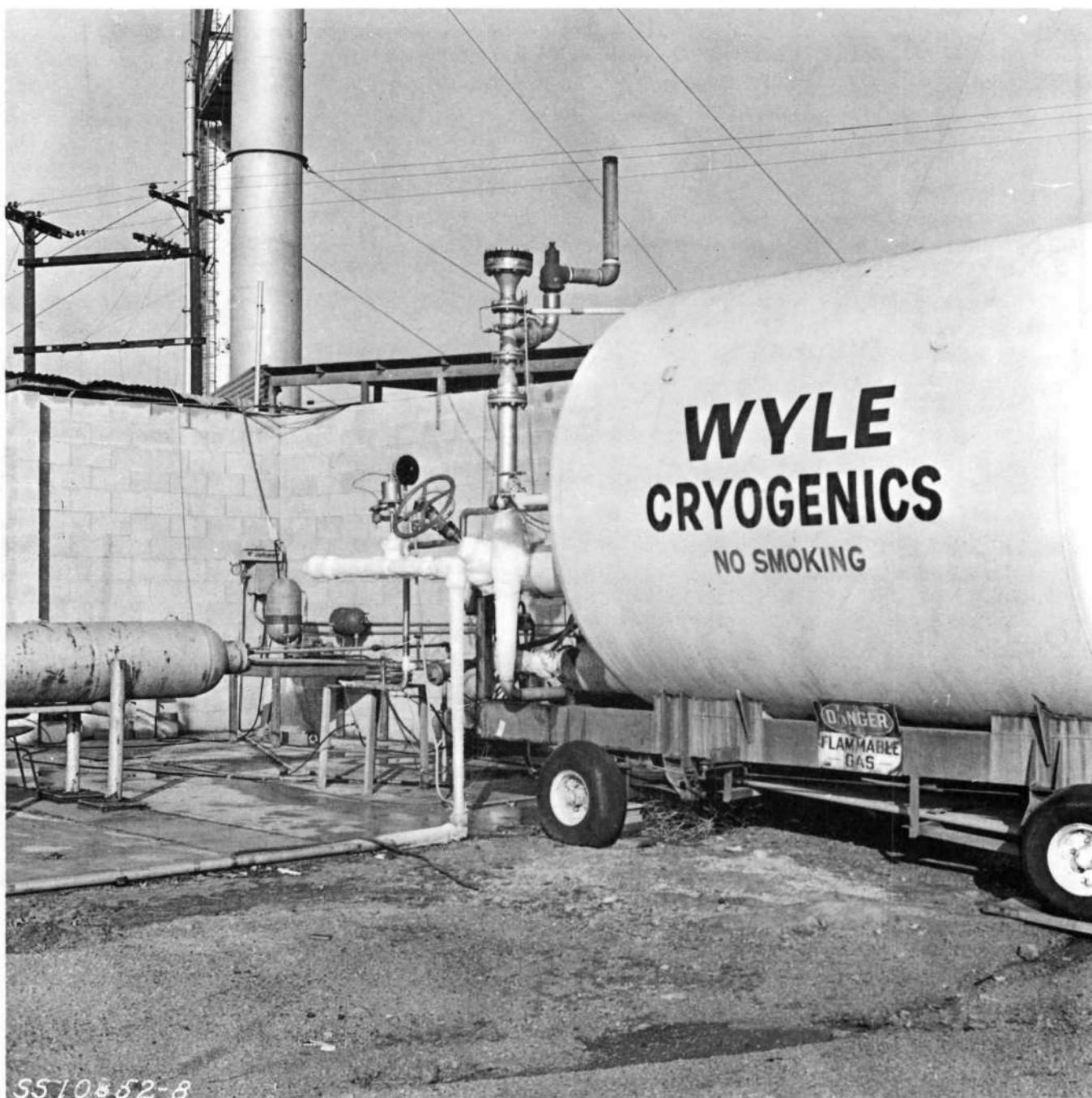
FLOW SYSTEM SCHEMATIC OF TEST SET-UP

FIGURE 18



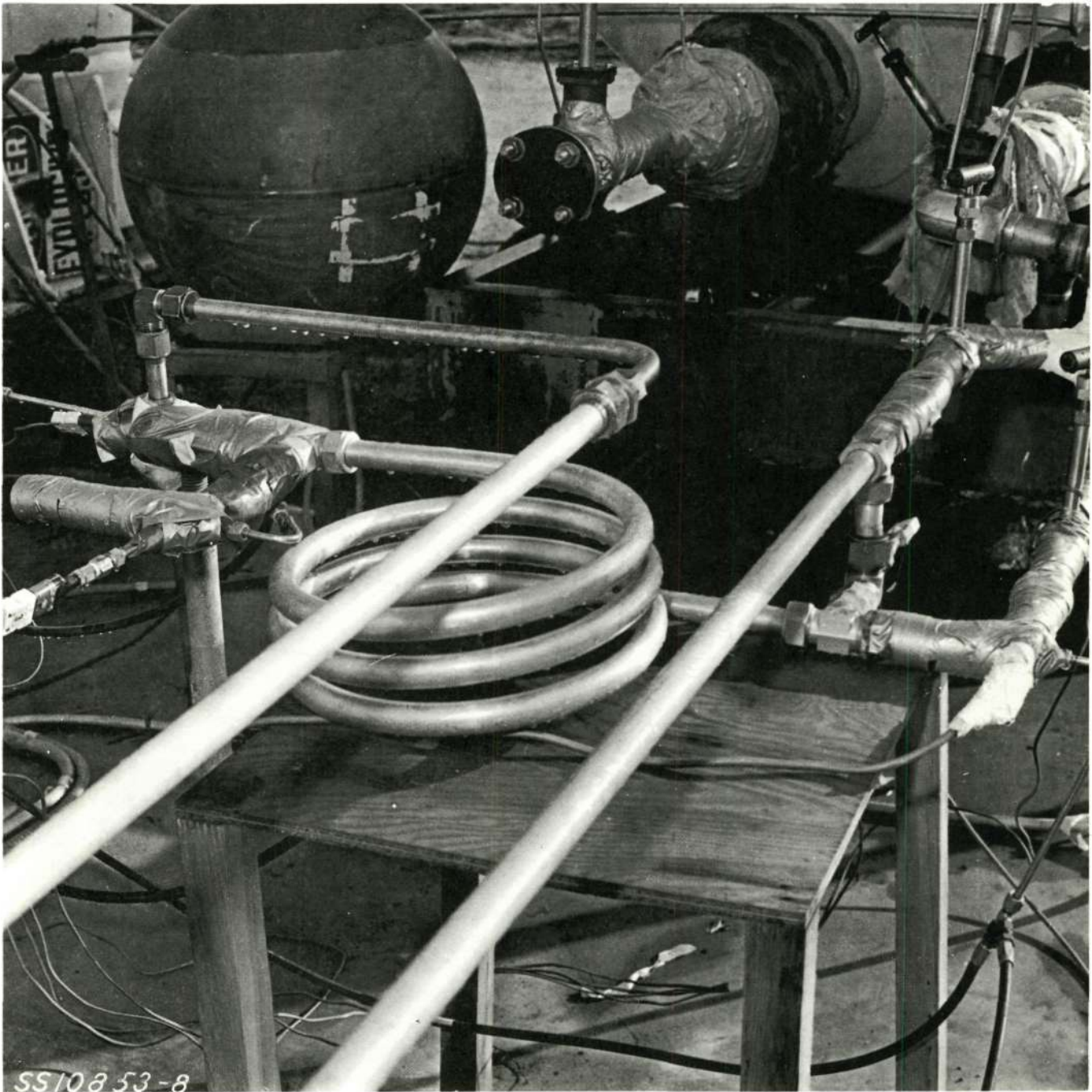
Test Setup

FIGURE 19



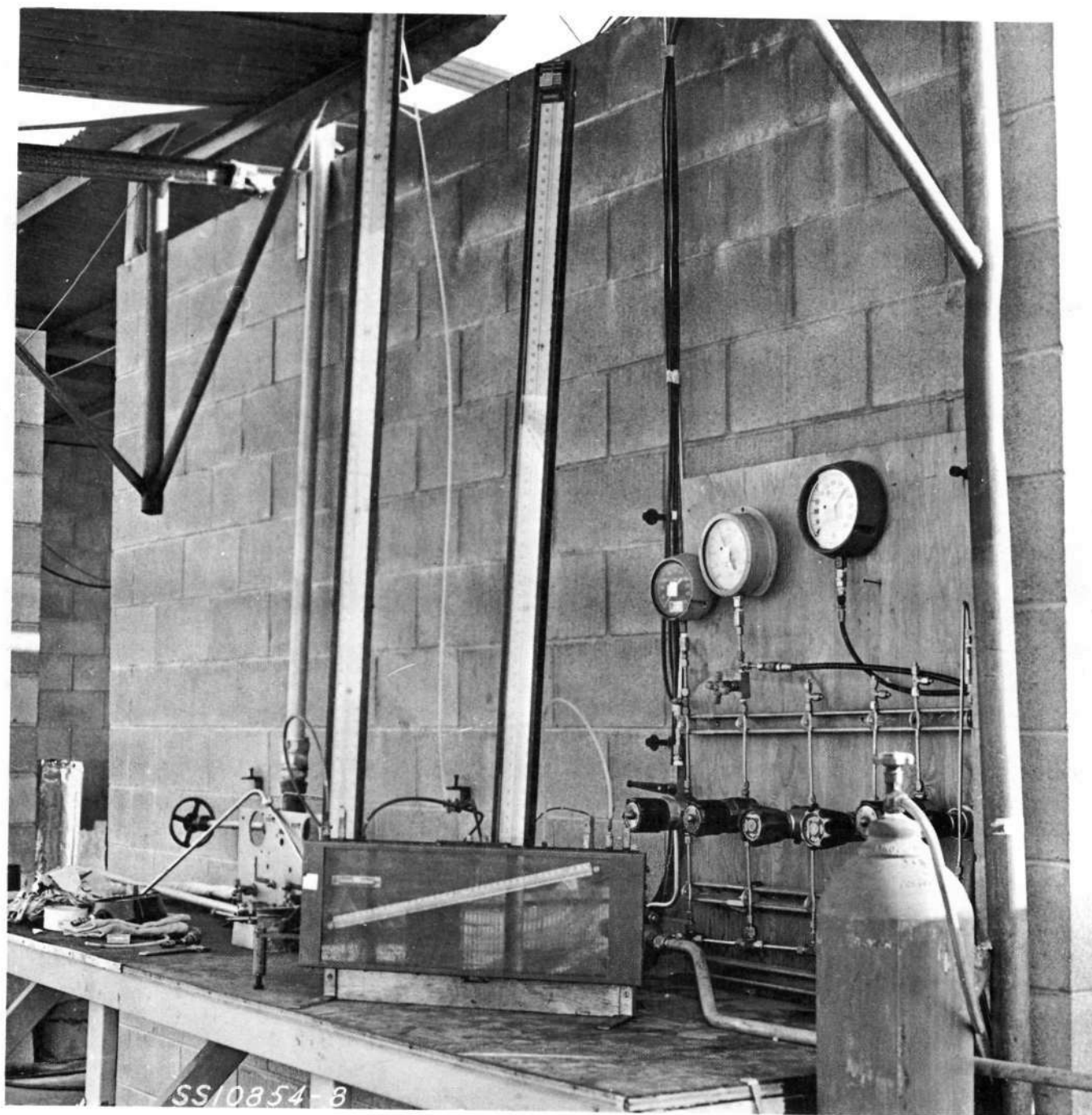
2000 Gallon - LH_2 Storage

FIGURE 20



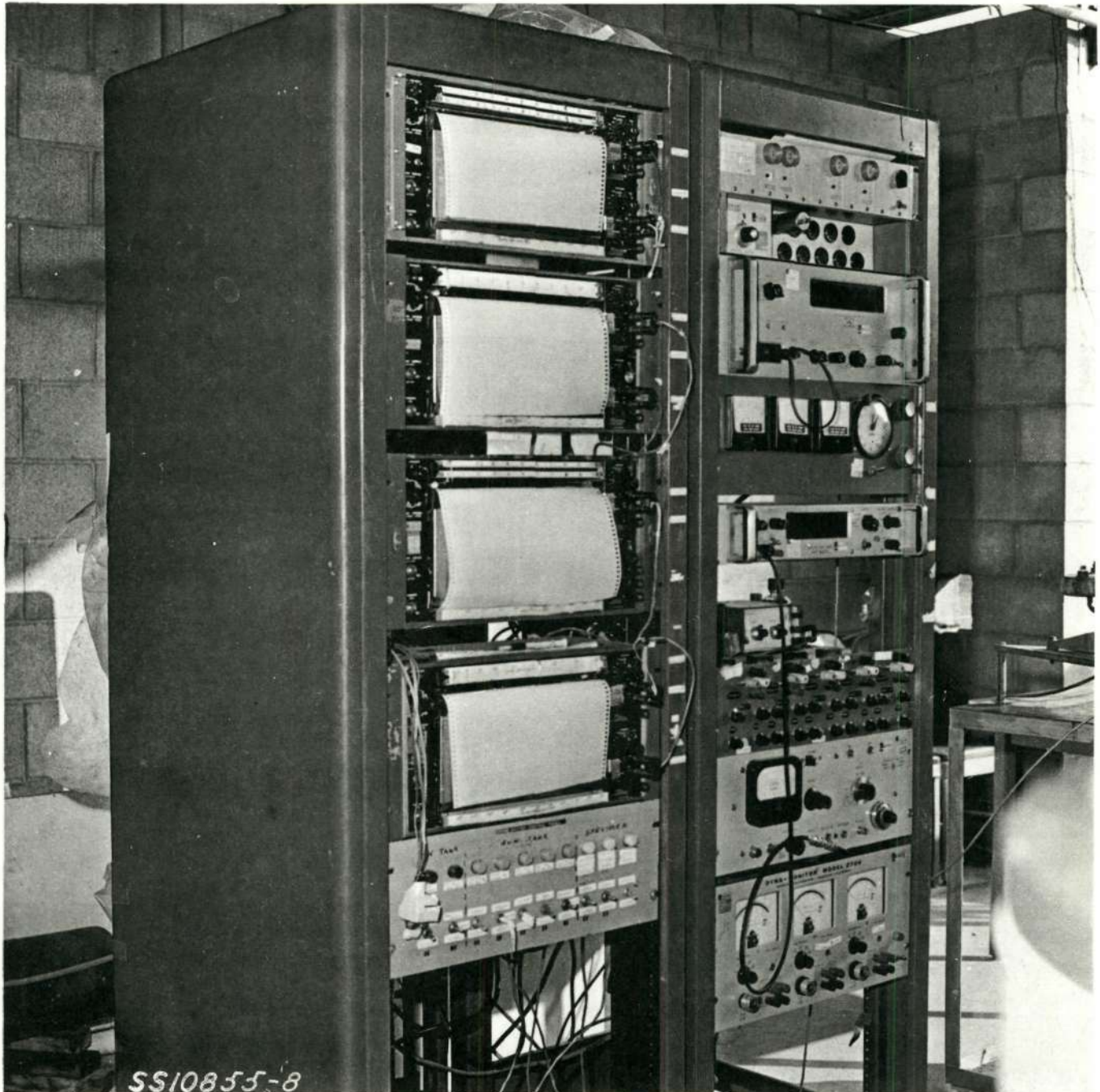
Test Specimen

FIGURE 21



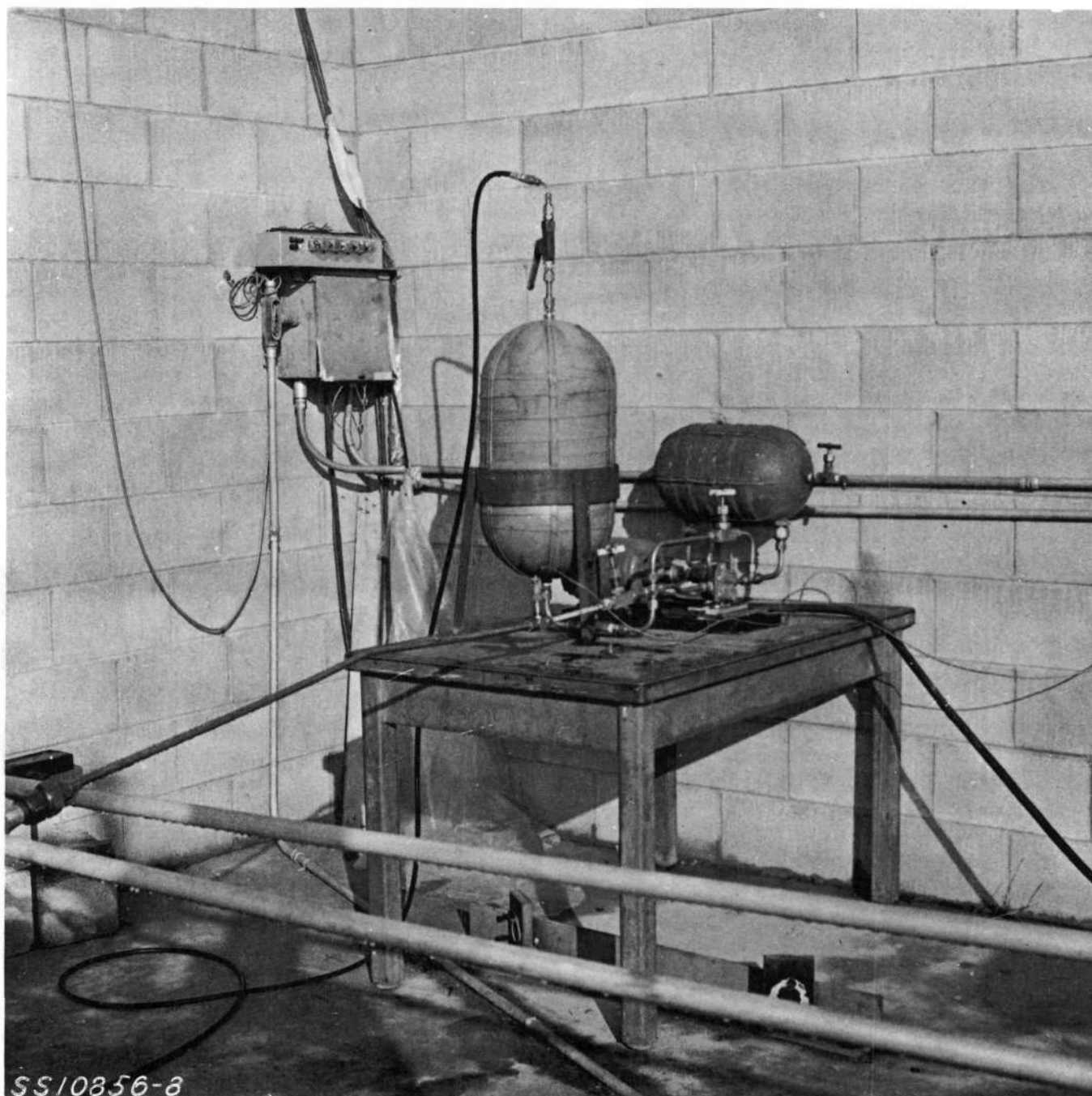
Instrumentation - Hydrogen Flow

FIGURE 22



Strip Chart Recorder

FIGURE 23



Freon Pump

FIGURE 24

Hydrogen inlet temperature was sensed with a Rosemount platinum probe and all other temperatures were sensed by calibrated copper constantan thermocouples. Hydrogen gas flow was measured using a Daniel's orifice as shown in figure 22. Hydrogen pressure and flow were controlled by manual valves upstream and downstream of the specimen. Hydrogen gas was discharged to a stack. Figures 19 and 21 show the specimen installed in the test rig. Note that liquid droplets on the specimen and adjacent tubing were caused by a rain shower preceding photography, and were not present during testing.

Freon 21 was circulated in a closed loop pressurized with nitrogen in order to insure that the Freon remained liquid. The Freon pump and ullage tanks are shown in figure 24. Flow rate was controlled by throttling valves and a pump bypass valve and was measured using a turbine flowmeter and recorded on a 4 channel recorder. The specimen was supplied with Freon 21 in a counterflow direction through one or both hot and cold heat exchangers via a mixing valve. Positioning the valve permitted assumption of the various heat loads as well as allowing specimen inlet temperature transients. Inlet and outlet Freon pressures were measured on 200 psi gages. Inlet and outlet temperatures were sensed with calibrated copper constantan thermocouples and were recorded on a Mosely recorder. Differential pressure across the Freon side of the specimen was sensed by a pressure transducer and also recorded on a Mosely recorder. A detailed list of instrumentation is contained in Appendix B.

Conduct of Test

In conducting the test the Freon loop was pressurized with dry nitrogen to the top of the inlet ullage tank. The Freon pump was then started and Freon flow and specimen inlet temperatures were adjusted. Hydrogen flow was next initiated through the dummy specimen until a minimum inlet temperature was achieved, thereupon the entire flow was diverted through the specimen. Steady state readings were taken when Freon and hydrogen flows and temperatures were stabilized. Note that inherent specimen instability caused oscillation of hydrogen conditions so average readings were taken. Testing was conducted in the following sequence as described in the test plan contained in Appendix A.

1. Steady state 5000 Btu/hr.
2. Freon temperature ramp change from minimum to maximum.
3. Freon temperature step change from minimum to maximum.
4. Shutdown and startup of hydrogen at 5000 Btu/hr load.
5. Steady state 20,000 Btu/hr.

6. Steady state 35,000 Btu/hr.
7. Steady state 50,000 Btu/hr.
8. Hydrogen flow shutoff at 50,000 Btu/hr.
9. Hydrogen flow startup at 50,000 Btu/hr.

STEADY STATE PERFORMANCE

Steady state performance tests were made on a tube - in - tube heat exchanger sized for the projected Shuttle application in order to verify the computerized analytical design procedures. In the tests Freon flow rate and temperatures were set at values approximating those expected in Shuttle and the flow rate of cryogenic H_2 at supercritical pressure was set to obtain nominal heat rejection rates of 5000, 20,000, 35,000 and 50,000 Btu/hr. A summary tabulation of the steady state data is presented in Table II. Two or three data points were taken at each nominal heat load and appear consistent.

At heat loads of 35,000 and 50,000 Btu/hr oscillations in H_2 flow and pressure on the order of $\pm 10\%$ of total values were experienced. A description of the oscillations, together with a discussion of their possible impact on the Shuttle system, is presented following the transient test results. Since an average value for fluctuating H_2 pressures, inlet temperature and flow had to be estimated to evaluate steady state conditions, this added somewhat to normal experimental uncertainties.

The percent error in the heat balance obtained between Freon and H_2 is listed for each data point in Table II. Excluding the 5000 Btu/hr points, the maximum percentage heat balance error was 7.7% at a 50,000 Btu/hr point, and consequently within acceptable experimental error. The three 5000 Btu/hr points have considerably higher percentage error, but because the Freon inlet to outlet temperature difference was small at this heat load an absolute error less than $2^\circ R$ in a Freon temperature reading would account for the error at two of the data points. It appears there may have been a systematic heat balance shift between different heat loads, illustrated in figure 25, which has not been explained. Heat loss or gain to the Freon through the outer heat exchanger tube wall would produce a shift in the indicated direction but is calculated to be relatively small.

The values for flow rates, pressures and inlet temperatures, recorded during testing for one data point at each nominal heat load were input to the computerized heat exchanger model. The resultant predictions at these conditions are listed in Table II. The predicted pressure drops are for the heat exchanger tubes only. Consequently the experimental pressure drops which include fittings are higher. The experimental Freon ΔP was erratic throughout the testing and is not believed to be precise.

The agreement between experimental and analytical values of the primary variable to be controlled by the heat exchanger, the Freon outlet temperature, is seen to be excellent. The maximum difference is $2.1^\circ R$ at the 5000 Btu/hr

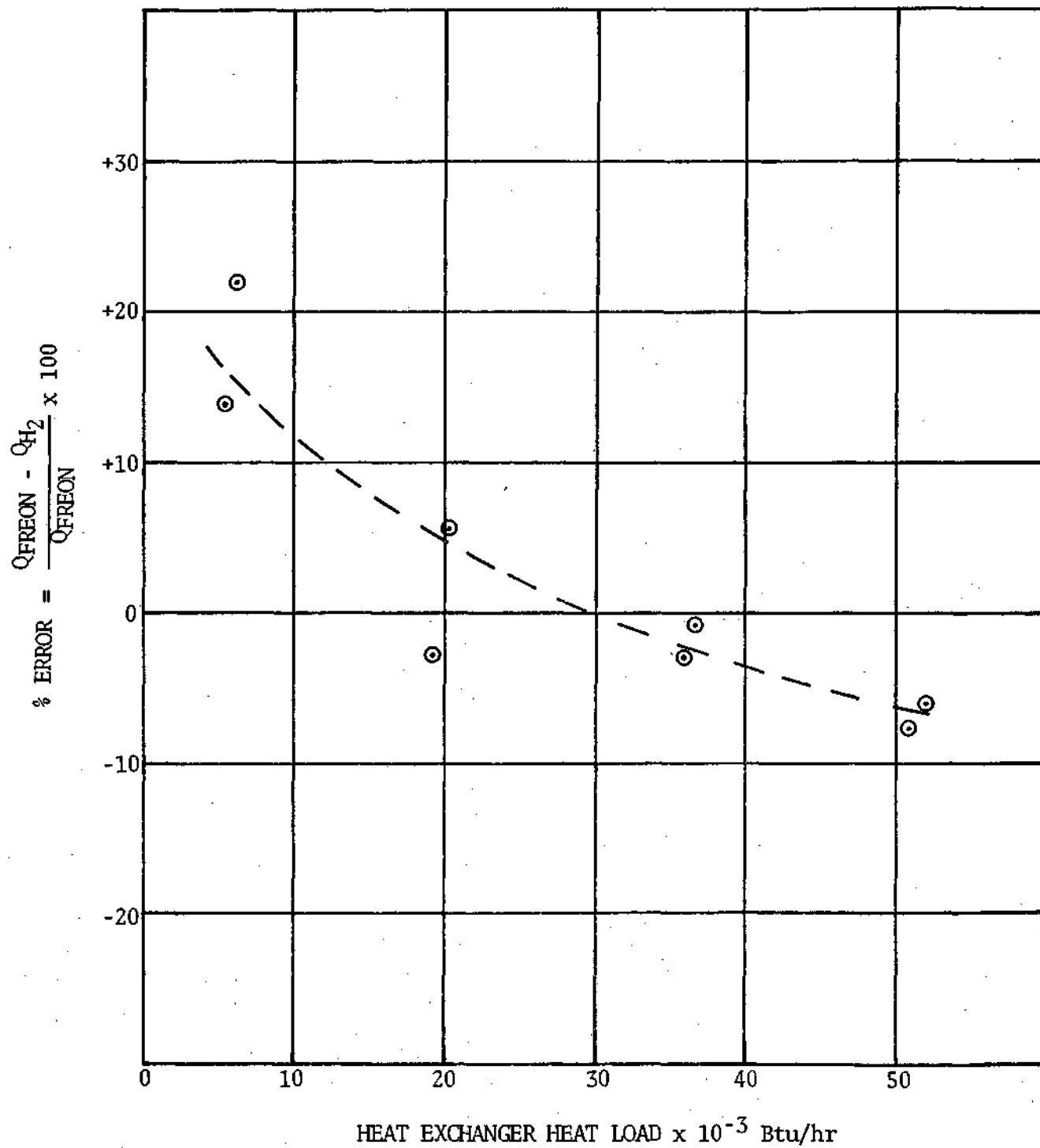
TABLE II

SUMMARY OF STEADY STATE TEST DATA AND ANALYTICAL PREDICTIONS

TEST NO.	TEST DATA												ANALYTICAL PREDICTIONS					FREON ΔT CORRELATION ERROR, % ②
	FREON 21						HYDROGEN						HEAT BALANCE ERROR, % ①	OUTLET FREON TEMP, °R	OUTLET H ₂ TEMP, °R	FREON ΔP , psi	H ₂ ΔP , psi	
	FLOW RATE, lb/hr	INLET TEMP, °R	OUTLET TEMP, °R	CALC. Q, Btu/hr	INLET PRESS, psia	PRESS. DROP, psi	FLOW RATE, lb/hr	INLET TEMP, °R	OUTLET TEMP, °R	INLET PRESS, psia	CALC. Q, Btu/hr	PRESS. DROP, psi						
1310	1708	507.8	494.9	5,460	114	4.7	3.00	71.0	509.5	205	4,710	.01	+13.7	496.9	502.1	2.6	.001	-14.7
1315	1710	507.2	494.8	6,100	114	4.7	3.02	70.5	510.0	205	4,760	.01	+22.0					
1342	1700	514.0	501.0	5,520	115	4.9	2.98	70.8	516.5	210	4,760	.01	+13.8					
1358	1672	552.4	507.6	19,120	114	4.7	11.50	62.0	527.9	210	19,660	.04	- 2.8	507.4	517.4	2.5	.01	- .45
1407	1676	550.5	501.2	20,450	112	4.7	11.40	62.0	524.0	210	19,320	.04	+ 5.5					
1439	1673	589.0	505.5	36,000	118	4.7	20.35	55.8	530.2	271	37,100	.03	- 3.0					
1445	1671	589.0	506.0	36,800	116	4.8	20.38	56.0	531.0	273	37,080	.04	- 0.8	506.7	505.1	2.6	.02	- 1.1
1457	1736	613.4	498.0	52,000	115	6.9	30.38	53.0	527.0	260	55,100	.07	- 6.0					
1504	1690	612.6	497.2	50,800	115	6.9	30.15	53.0	526.6	260	54,700	.08	- 7.7	497.6	485.5	2.7	.03	+ .35

$$\textcircled{1} \text{ \% heat balance error} = \frac{Q_{\text{Freon}} - Q_{\text{H}_2}}{Q_{\text{Freon}}} \times 100$$

$$\textcircled{2} \text{ \% Freon } \Delta T \text{ correlation error} = \frac{\Delta T_{\text{test}} - \Delta T_{\text{analytical}}}{\Delta T_{\text{test}}} \times 100$$



STEADY-STATE TEST POINT HEAT BALANCE ERROR

FIGURE 25

point, which represents a 14.7% difference between the relatively small experimental and predicted Freon ΔT in the heat exchanger. At the three other heat loads the differences between the experimental and analytical Freon ΔT are very small percentage wise. The experimental heat imbalance appears primarily as a difference between experimental and predicted H_2 outlet temperature, suggesting the imbalance results from H_2 heat capacity or temperature readings. These could be the result of hydrogen oscillations at the higher heat loads.

TRANSIENT TEST RESULTS

The cryogenic heat exchanger would be employed for heat rejection from the Shuttle Freon loop in such a way that a change in the system heat load would result in a change of the Freon temperature entering the heat exchanger. The heat exchanger control must then respond to the changed heat load by changing H_2 flow to the heat exchanger so as to maintain the Freon exit temperature within a relatively narrow control band.

In order to design an appropriate control the "open loop" response of the Freon exit temperature to a change in both Freon inlet temperature and H_2 flow rate, as determined by the thermal dynamics of the heat exchanger itself, must be predicted. An analytical model of the heat exchanger dynamics was programmed for a computer for the purpose of control analysis. This model was described in NASA CR 115569 and is included in the dynamic model of the Shuttle Freon loop described elsewhere in this report. The primary purpose of the transient testing reported below was to verify the accuracy of this analytical model.

Tests were performed in which there was:

1. An independent change in H_2 flow into the heat exchanger (with Freon inlet temperature and other variables held constant⁽¹⁾), and
2. An independent change in Freon temperature at the inlet to the heat exchanger.

(1) Due to practical limitations in the test setup, it was not possible always to maintain exactly constant values of variables other than the one deliberately changed. For instance, a variation in Freon flow resulted from changing the Freon inlet temperature because the mixing valve for Freon from two different temperature reservoirs was not linear. However, such undesired changes were small enough that they were adequately compensated for in correlating the data by using average values over the period of the transient.

The resultant change in Freon exit temperature as a function of time was recorded (along with other significant variables) and compared to predictions obtained from the computerized model. Although with a control system in operation a Freon inlet temperature change would generally be accompanied by a simultaneous change in H_2 flow rate, it is desirable from an analytic viewpoint to study each separately as was done in the testing because the time response of the Freon exit temperature is significantly different in the two cases.

Freon Exit Temperature Response to H_2 Flow Changes

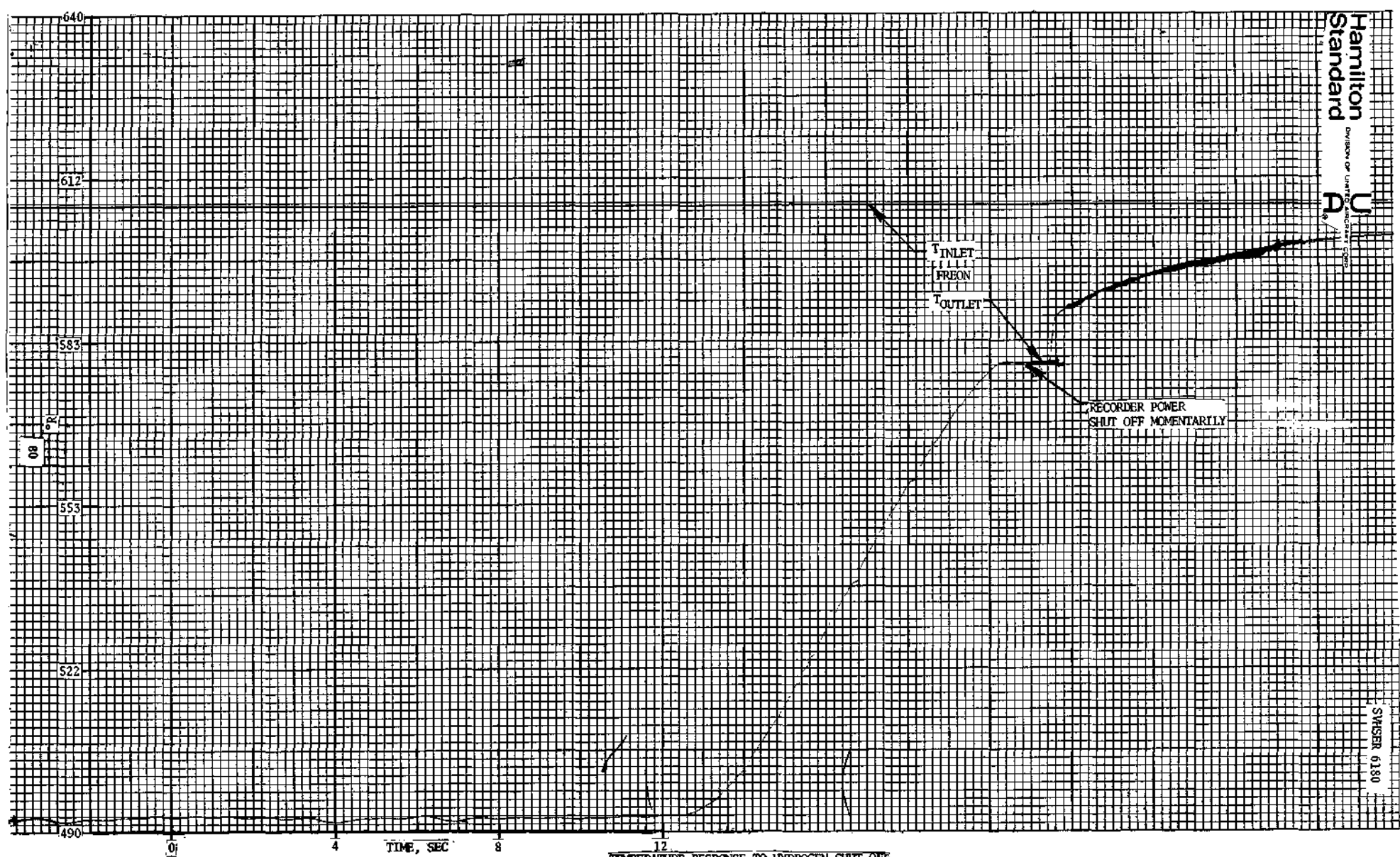
Figure 26 shows the recorded experimental response to shutting off H_2 flow at a beginning nominal heat load of 50,000 Btu/hr. This data has been plotted in Figure 27. Figure 28 is a plot of Freon exit temperature response data when H_2 was first turned off, then turned on at a nominal heat load of 5000 Btu/hr. In the tests, H_2 flow to the heat exchanger was regulated by a hand valve and an off or on "step" change in H_2 flow took place over the finite time required to turn the valve. Inspection of the data indicates this time was on the order of 2 or 3 seconds.

Prediction of these Freon exit temperature transients obtained from the computerized model are also plotted in figures 27 and 28. An instantaneous step change in H_2 flow was input to the computer, so that the analytical line leads the data by about two seconds and does not show the rounding at the beginning of the transient produced by closing the valve over a period of 2 or 3 seconds. However, the slope of the analytical Freon exit temperature response very closely matches the data. This may be seen most clearly in figure 27 where the transient extends over a sufficient range for the slope to be apparent.

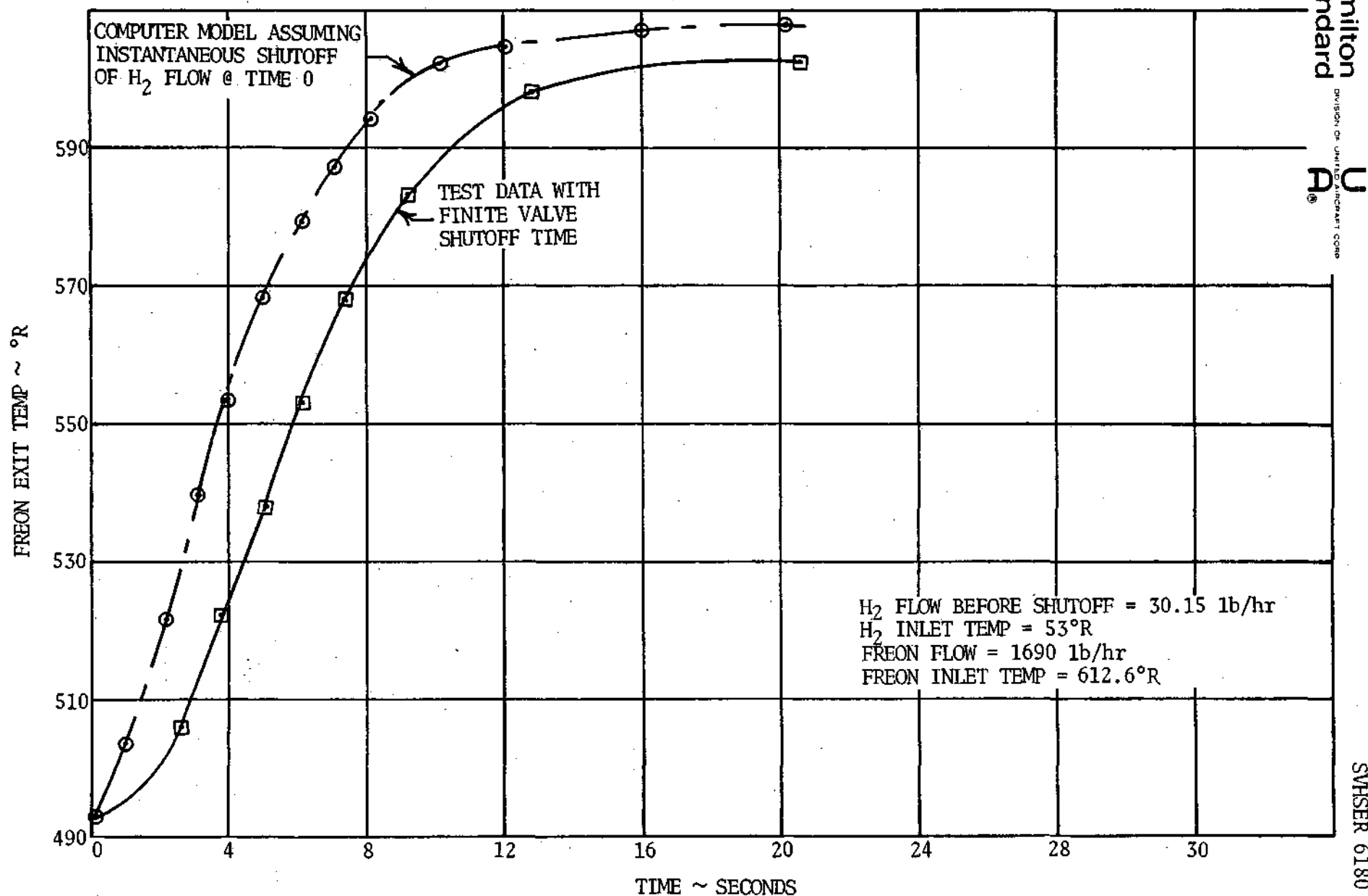
The nature of the response of the Freon exit temperature to a step change in H_2 flow may be seen in these plots. A change in Freon exit temperature begins almost immediately after H_2 flow is changed, and exhibits a nearly constant slope over the greater portion of the transient. These two characteristics result from the fact that the time constant of the heat exchanger wall separating the H_2 and Freon is small, both absolutely and relative to the transport time of the Freon in the heat exchanger.

Freon Exit Temperature Response to Freon Inlet Temperature Change

Figure 29 is a reproduction of all the recorded variables during a transient initiated by a change in Freon inlet temperature. In figure 30 the inlet temperature change and the outlet temperature response have been plotted from the data. It was possible to input to the computerized model a Freon inlet temperature change very close to that experimentally recorded. As a consequence, the predicted exit temperature response plotted in figure 30 closely matches the data throughout the entire transient (unlike the H_2 flow change transients where the initial H_2 flow change could not

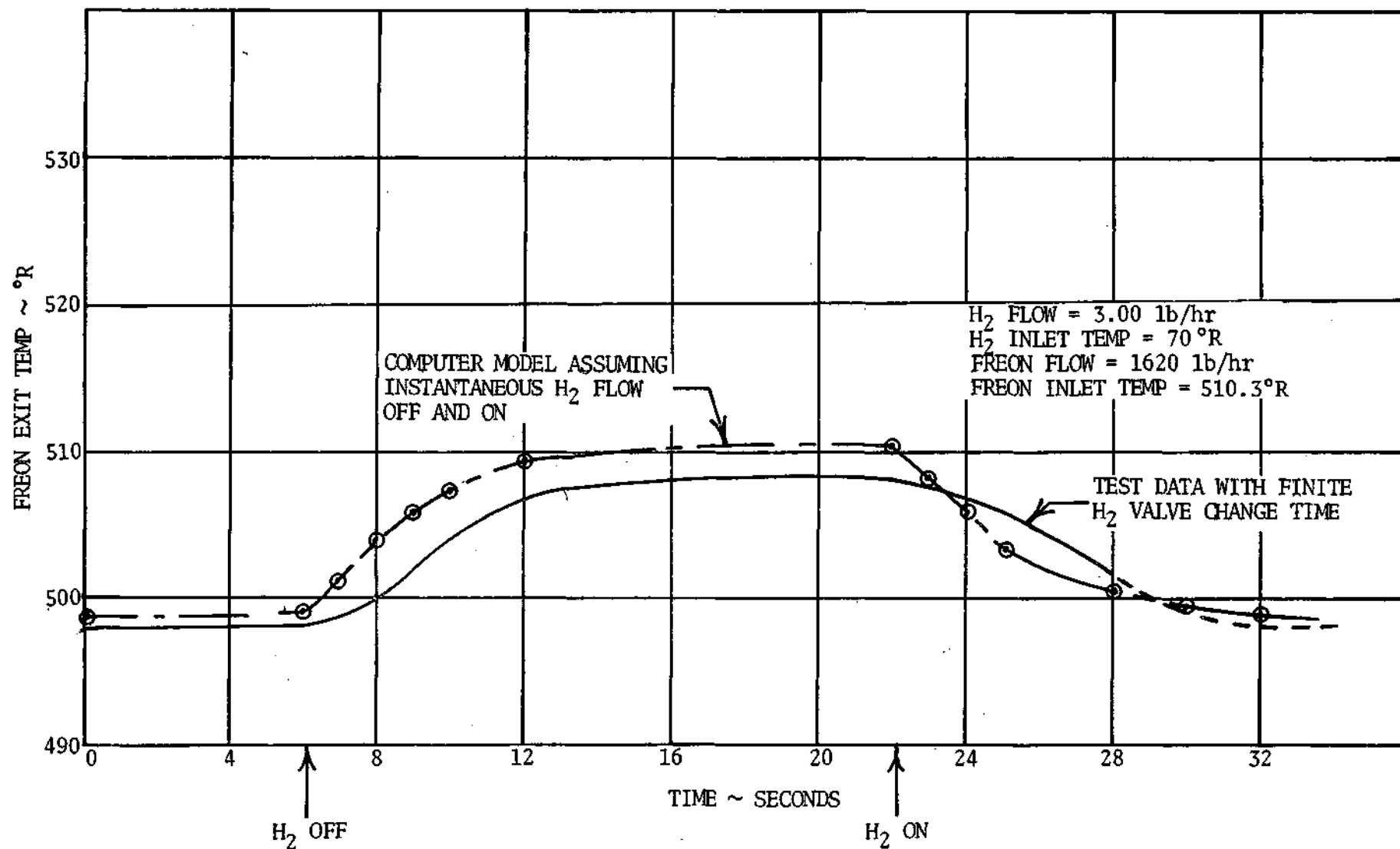


TEMPERATURE RESPONSE TO HYDROGEN SHUT OFF
FIGURE 26



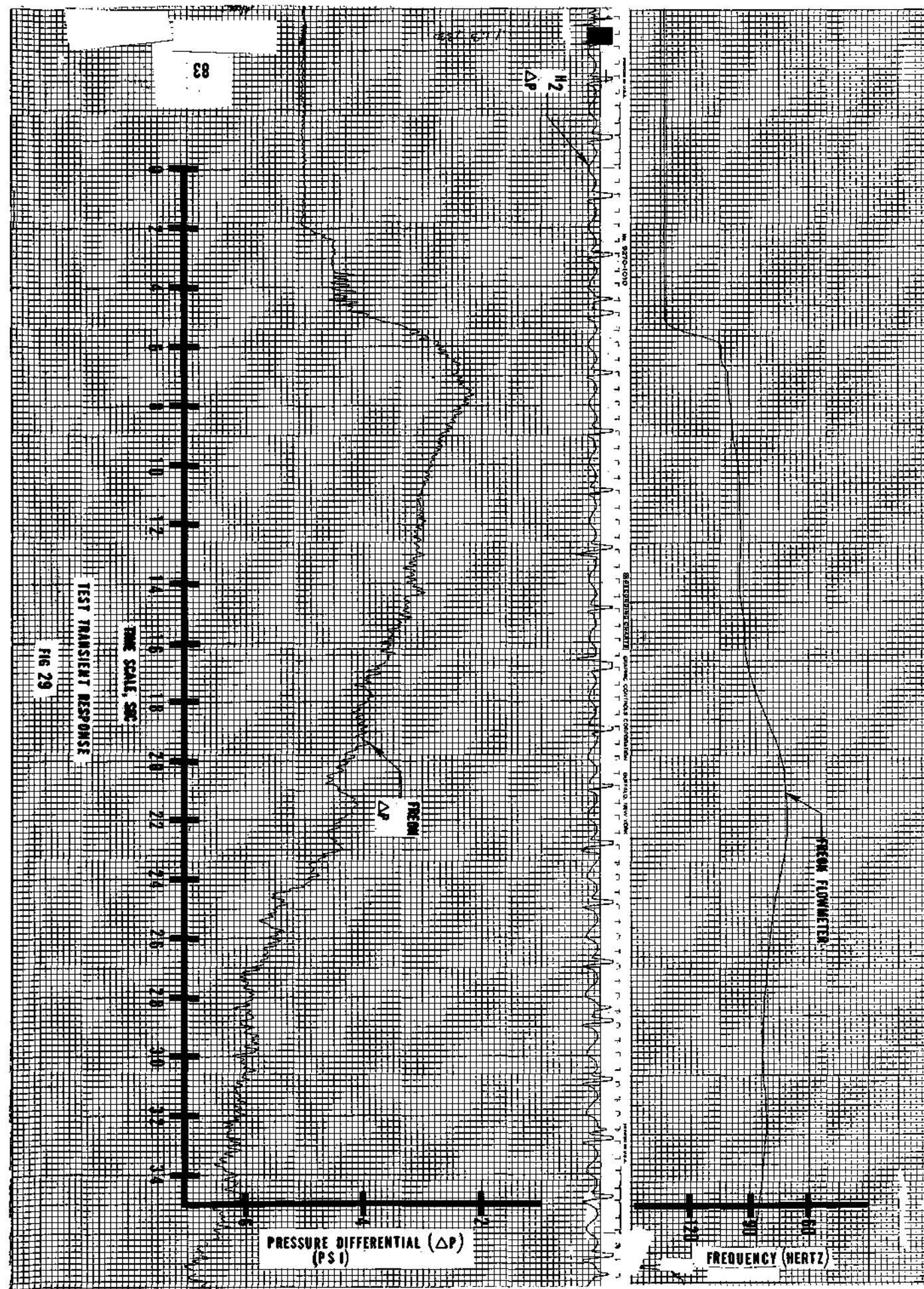
RESPONSE OF FREON TEMPERATURE AT EXIT TO HEAT EXCHANGER
 TO SHUTTING OFF H_2 FLOW TO HEAT EXCHANGER
 (NOMINAL 50,000 Btu/hr LOAD)

FIGURE 27

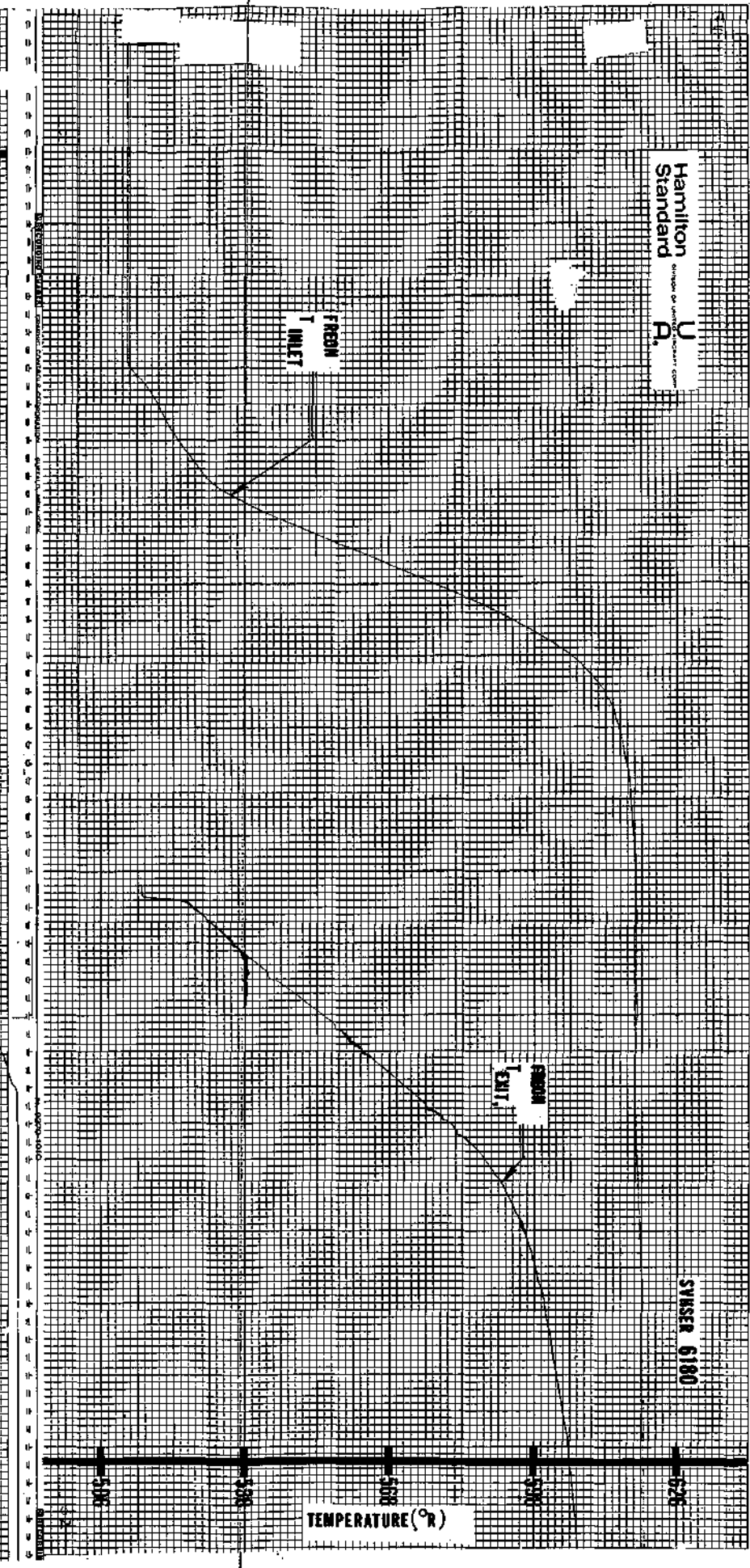
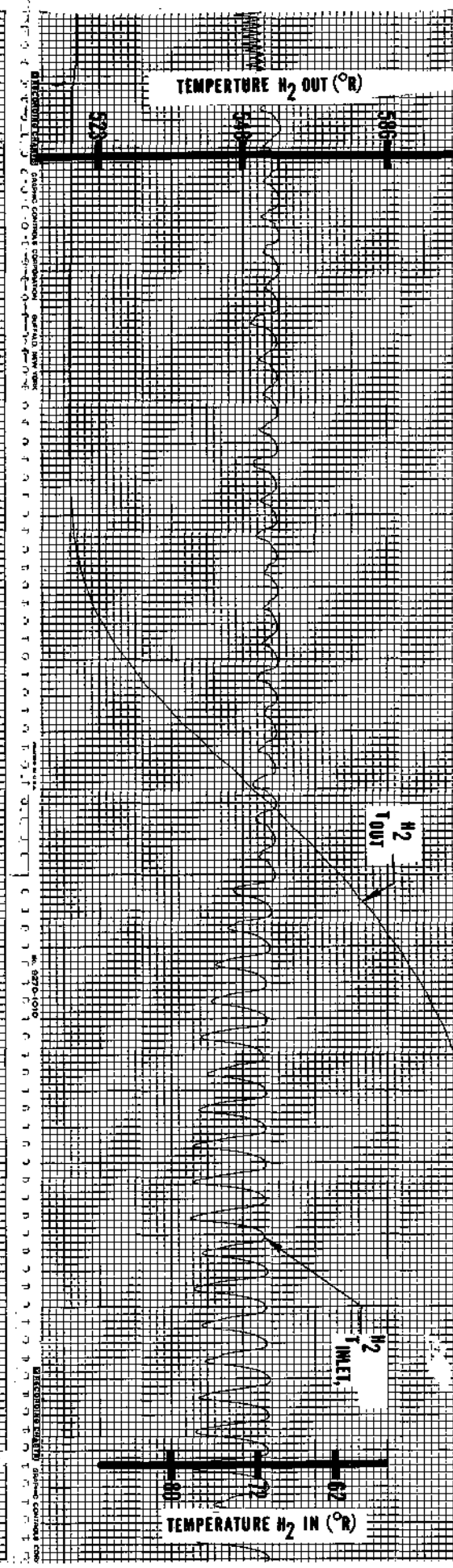


RESPONSE OF FREON TEMPERATURE AT EXIT TO HEAT EXCHANGER
TO SHUTTING OFF AND STARTING H₂ FLOW
(NOMINAL 5,000 Btu/hr LOAD)

FIGURE 28

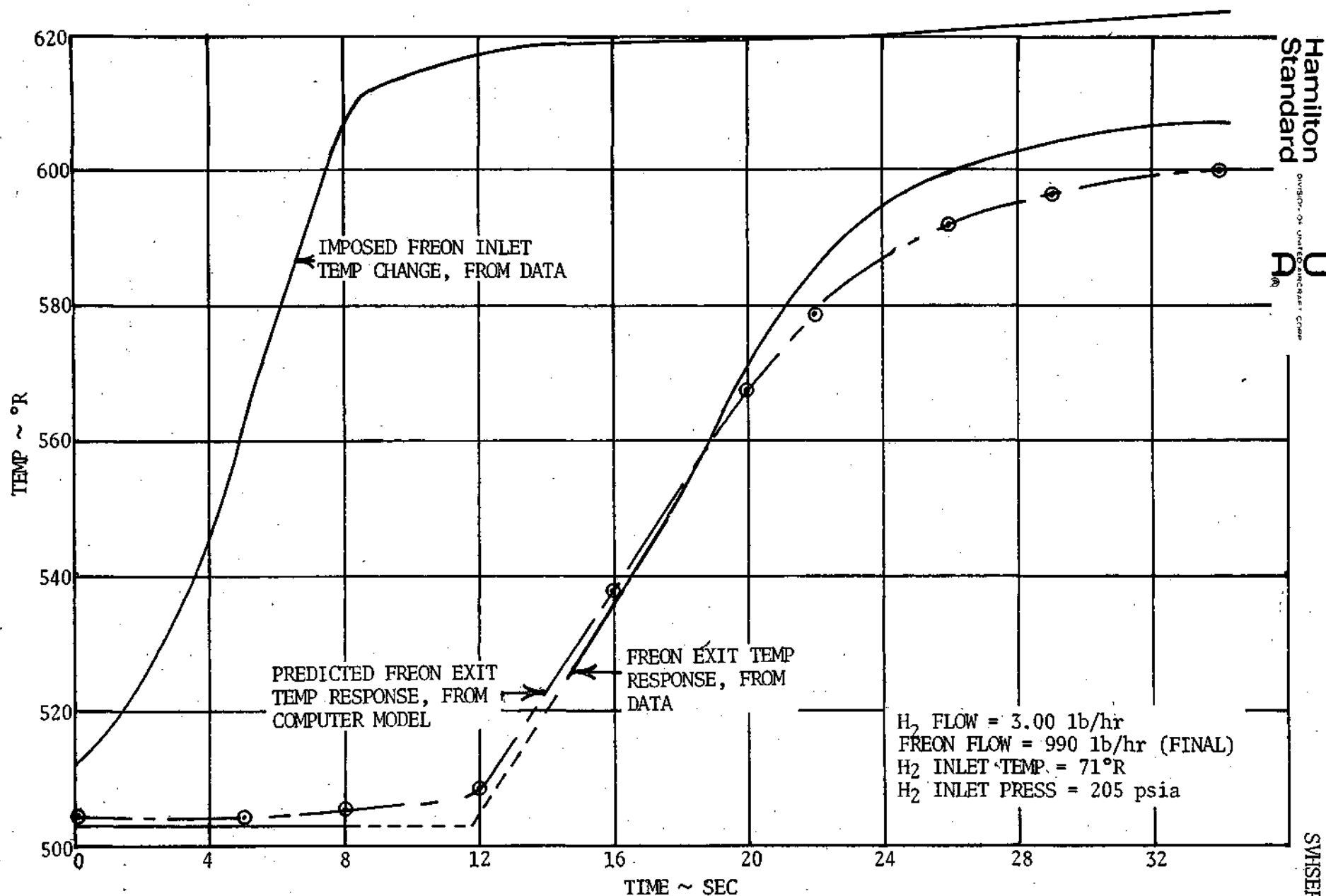


FOLDOUT FRAME 1



FOLDOUT FRAME 2

83/84



RESPONSE OF FREON OUTLET TEMPERATURE TO CHANGE
 IN INLET TEMPERATURE
 (NOMINAL 5,000 Btu/hr LOAD)
 FIGURE 30

be exactly input). The slight divergence at the end of the transient shown in figure 20 is caused by a shift in Freon flow rate.

The difference between the Freon exit temperature response to a change in H_2 flow and a change in Freon inlet temperature may be seen by comparing figures 27 and 30. The delay in response shown in figure 30 is due in large part to the time it takes a particle of Freon to traverse the heat exchanger.

DISCUSSION OF INSTABILITIES ENCOUNTERED DURING TESTING

Flow and pressure instabilities have been widely documented when cryogenic and non-cryogenic fluids are vaporized under forced convection, and when supercritical fluids are heated. Two-phase flow in a fluid being vaporized in a heat exchanger is always unstable to some degree (if only on a microscopic level) because interrelations between heat transfer and fluid momentum are typically such that local perturbations in the density of the vaporizing fluid occur. The situation when a supercritical fluid is heated is analogous because gradients in fluid properties, both across the boundary layer and in the direction of flow, are similar to those in a vaporizing liquid. Under certain circumstances flow and pressure variations in the heat exchanger become oscillatory and large in magnitude. Hypotheses concerning the "feedback" mechanisms which produce these oscillations will be discussed briefly further on. Such oscillations in pressure and/or flow are detrimental if undesirable effects are produced in the system of which the heat exchanger is a part.

During the testing covered by this report the instrumentation employed was not designed to completely define the oscillations. For instance, the phase relations between inlet and outlet pressure variations were not continuously recorded and would be of value. However, the following general description of the oscillations may be given.

The maximum pressure and flow oscillations encountered were on the order of $\pm 10\%$ of total values and were at a frequency of about one hertz. The oscillations had a limit-cycle nature: that is, they did not increase indefinitely but remained at the same amplitude. The largest amplitude appeared at the highest heat load ($\approx 50,000$ Btu/hr), while at the lowest load (≈ 5000 Btu/hr) none were detectable on total pressure or flow gages although variations appeared on the H_2 heat exchanger ΔP transducer and the H_2 inlet thermocouples. Flow variations produced variations in the thermocouple reading H_2 inlet temperature, but often none in the H_2 outlet thermocouple and never any significant variations in thermocouples reading Freon temperatures. Variations in H_2 inlet temperature were nearly linear (i.e., sinusoidal) while those appearing on the ΔP transducer were not. Recorded oscillations in H_2 inlet temperature and ΔP may be seen in figure 29.

The effect of such oscillations as were encountered during testing may be reasoned to be non-detrimental to the Shuttle system in which it is proposed to use the cryogenic heat exchanger for the following reasons:

1. The instability was not of sufficient severity to cause oscillations in the Freon outlet temperature. Consequently there is no possibility of unstable feedback through the heat exchanger control which senses and controls Freon outlet temperature by modulating H_2 flow; the controls would not "know the H_2 flow was oscillating."
2. The experimental steady-state and transient thermal performance of the heat exchanger appeared not to be affected by the oscillating flow, and is analytically predictable.
3. The Shuttle H_2 feed system proposed for the cryogenic heat exchanger consists of a large storage tank supplying H_2 to fuel cells. H_2 is to be returned from the cryogenic heat exchanger to the tank and the heat exchanger serves the supplementary function of pressurizing the tank by adding thermal energy. This large tank may be expected to be an efficient damper for pressure oscillations originating in the cryogenic heat exchanger, so no detrimental effects on tank pressure control are expected. A "hard" H_2 circulating pump, i.e., one with steep pressure versus flow characteristics, would prevent pressure signal feedback between the tank inlet and outlet and therefore prevent a possible resonant condition within the tank.

Although no detrimental system effects are expected from such oscillations as were encountered in the testing reported, further definition of the nature of the instabilities encountered is clearly required before such a system could be designed with confidence, for at least two reasons:

1. The system itself has not been completely defined and is subject to change as Shuttle design progresses.
2. Since the tests were designed only to obtain steady state and transient thermal performance of the heat exchanger, the interface between the heat exchanger and its H_2 feed and exhaust systems were not designed to simulate an indefinite H_2 system. H_2 pressure drop across the inlet valve was much larger than in the contemplated system, while a valve having choked flow at the heat exchanger exit used in the tests would not be part of the system. (The effect of the large inlet pressure drop would be to decrease the magnitude of pressure fluctuations in the system while the choked orifice would be expected to increase their magnitude.) In the absence of an accepted general theory concerning the mechanisms underlying instability in supercritical heat exchangers, data cannot be extrapolated with confidence to other conditions.

A summary of literature on experimental and analytical investigations into instabilities in vaporizing and supercritical heat exchangers is presented below to provide background. The discussion is then concluded with recommendations for further investigations aimed at the Shuttle application. It should be noted at this point that even without a cryogenic heat exchanger rejecting heat from the Freon coolant loop to the external pressurization loop for a fuel cell H_2 tank, instability may be encountered in this external pressurization loop when heat is added by an electric heater for the purpose of tank pressure control.

The common starting points for formulating an analytical model of oscillations in vaporizing and supercritical heat exchangers is of course obvious:

Write the partial differential equations for fluid continuity and momentum, and for energy transfer to the fluid in the heat exchanger. A problem arises in the region where the fluid is vaporizing or supercritical because density gradients and gradients in other fluid properties occur in a direction perpendicular to flow due to heat addition. Some of the known potentially de-stabilizing factors in two-phase flow with heat addition, such as slip between phases in the direction of flow and a non-linear dependence of heat transfer on the wall to fluid (bulk) temperature difference are difficult to investigate experimentally and exact theoretical relationships to other variables have not been established.

Because of the complexities of the situation, some of which are described above, an exact solution to the differential equations is not possible: extreme simplifications are required to obtain a manageable solution. Depending on which features are retained in a simplified model somewhat different pictures of the mechanisms underlying oscillations are obtained. For example, the oscillatory storage of fluid mass in the heat exchanger that must occur for flow to oscillate may be envisioned as:

1. Due to variations in the width of a liquid or heavy supercritical fluid core riding on a vapor film created by heat addition at the wall (6);
2. due to the piston like movement of such a core; or
3. due to fluid density waves traversing the heat exchanger (4) (7).

The three pictures need not be exclusive.

A commonly employed simplification is to linearize the differential equations so that Laplace transforms may be taken and stability criteria directly applied. Freidly and Krishnan (1) have recently applied Poincare's method to obtain a model providing limit cycle behavior, which the oscillations usually exhibit and which a linearized analysis cannot predict.

Critical parameters separating stable from unstable conditions which result from different models include:

1. Density ratio between the fluid at the heat exchanger inlet and outlet (4);
2. the ratio of heat flux to mass flow rate in the vaporizing or supercritical region (6); and
3. the ratio of fluid pressure drops in the system upstream and downstream of the point at which the principal density change may be thought of as occurring in the heat exchanger (e.g., the end of the vaporizing section) (2).

Much more experimental work is required to determine which simplifications can be most fruitfully employed to construct a model which can be used as a tool for predicting instability in the Shuttle application. In the absence of such a tool, experimental data collected under conditions most similar to the one being considered should prove most reliable. The most extensive data on oscillations resulting from heating supercritical H_2 in tubes has been taken by Thurston (6), who had success in correlating his own data using the parameters:

$$N_{BO} = \frac{\text{heat flux}}{\text{mass flow rate} \times \text{heat of vaporization}}$$

$$N_{SV} = \frac{\text{specific volume change due to vaporization}}{\text{specific volume of dense phase}}$$

At the inception of oscillations,

$$N_{BO} = .0045 (N_{SV})^{-.75}$$

This correlation must be applied with caution, however, when conditions are different. One difference between Thurston's tests and the tests reported here is that Thurston's tube was electrically heated, giving a different response of wall temperature to a change in the H_2 heat transfer coefficient than would occur with a fluid being cooled on the other side of the tube.

Further investigations, both experimental and analytical, into the nature of the mechanisms producing oscillations, such as were encountered in the tests reported, would clearly be of value as a design guide in cryogenic heat exchanger applications. The sort of design changes that might be made to stabilize the heat exchanger under the conditions tested are suggested by previous work. For example, using Thurston's criterion, stable H_2 flow could be obtained by insulating the tube wall in the supercritical region

and having the H_2 flow passage provided by a spiral insert inside the tube in the supercritical region. However, such measures cannot be taken with confidence without better understanding.

Such tests might be most productive if conducted with scaled-down heat exchangers. In spite of the obvious danger of scaling when the phenomenon is not completely understood and the difficulties in accurately instrumenting the tests in a miniature scale, the cost of employing the sophisticated instrumentation that could be required together with large H_2 flows would appear prohibitive.

APPENDIX A

TEST PLAN

BREADBOARD CRYOGENIC HEAT
EXCHANGER PROGRAM

TEST PLAN

PREPARED UNDER CONTRACT NAS 9-12725

BY

HAMILTON STANDARD
DIVISION OF UNITED AIRCRAFT CORPORATION
WINDSOR LOCKS, CONNECTICUT

FOR

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
MANNED SPACECRAFT CENTER
HOUSTON, TEXAS

OCTOBER 1972

Prepared By:


L. F. Desjardins
Program Engineer

Approved By:


F. H. Greenwood
Program Manager

1.0 GENERAL INFORMATION1.1 Scope

The purpose of this plan is to define the procedures and equipment to be utilized to test a tube - in - tube cryogenic heat exchanger. The objective of the test program is the correlation of analytically determined steady state and dynamic values with test results. Testing will consist of steady state runs for thermal performance and transient runs for evaluation of the transfer function. Performance will be evaluated at four different conditions and transient operating characteristics will be measured at five conditions. Wyle Laboratories of Norco, California will conduct all testing utilizing super critical hydrogen and Freon 21.

1.2 Applicable Documents

NASA/MSC Contract NAS 9-12725
Hamilton Standard Program Operating Plan B88-001
Tube - in - tube heat exchanger drawing SVSK 86021

1.3 Functional Requirements

The tube - in - tube counterflow heat exchanger is designed to operate | A
in the Space Shuttle Orbiter. Stored sub critical hydrogen provides
a nominal 50,000 Btu/hr of cooling. The heat exchanger shall be
capable of operation at heat loads as low as 5000 Btu/hr without
freezing.

2.0 TEST PLAN AND PROCEDURES

The heat exchanger will be subjected to steady state tests to determine thermal performance and transient tests for dynamic analysis.

2.1 Steady State Tests

These tests will be conducted at each of the following heat loads:

1. 5,000 Btu/hr
2. 20,000 Btu/hr
3. 35,000 Btu/hr
4. 50,000 Btu/hr

2.2 Transient Tests

The following transients will be imposed on the heat exchanger:

1. Freon inlet temperature change from 52°F - 160°F at a uniform rate of 20°F ± 5°F/min.
2. Freon inlet temperature change from 52°F - 160°F in a step change.
3. Start hydrogen flow @ 5,000 Btu/hr.
4. Start hydrogen flow @ 50,000 Btu/hr.
5. Stop hydrogen flow @ 50,000 Btu/hr.

2.3 Test Conditions

Conditions applicable to the above test runs are as follows:

HEAT LOAD	FREON			HYDROGEN		
	FLOW RATE	INLET TEMP.	OUTLET PRESS.	FLOW RATE	INLET TEMP.	INLET PRESS.
(Btu/hr)	(lb/hr)	(°F)	(psia)	(lb/hr)	(°F)	(psia)
5,000	1600	52	110 Min	3.0	-420	190 Min
20,000	1600	88	110 Min	12.0	-420	190 Min
35,000	1600	124	110 Min	21.0	-420	190 Min
50,000	1600	160	110 Min	30.5	-420	190 Min

A

2.4 Test Procedure

Definition of procedure, instrumentation and equipment are described in Wyle test procedure #3474 contained in Appendix A.

2.5 Data

The test data acquired during the steady state runs will be compared with prior performance calculations or will be recalculated if data is not coincident with specific data points. The data obtained from the transient runs will be input to the computer for determination of effect on accuracy and stability.

2.6 Test Schedule

Testing will be started within 2 - 4 weeks of approval and will last approximately 2 weeks.

TEST PROCEDURE NO. 3474DATE: 28 SEPTEMBER 1972

WYLE LABORATORIES / El Segundo, California

TEST PROCEDURE
FOR
HEAT TRANSFER TEST
ON
HEAT EXCHANGER
PART NUMBER SVSK86021
FOR
HAMILTON STANDARD

APPROVED BY: _____
FOR: _____

APPROVED BY: C. L. Chab
FOR: _____ WYLE LABORATORIES

APPROVED BY: _____
FOR: _____

APPROVED BY: W. R. Schubert
FOR: _____ WYLE LABORATORIES

APPROVED BY: _____
FOR: _____

PREPARED BY: W. B. Kelly
WYLE LABORATORIES

REV. NO.	DATE	PAGES AFFECTED	BY	APP'L.	DESCRIPTION OF CHANGES
#1	Oct. 1972	page 5	HS		Par. 4.1.1-changed freon inlet temps. and hydrogen flow rates.
		page 5	HS		Par. 4.1.2 52°F was 60°F.
		page 5	HS		Par. 4.1.3 52°F was 100°F
		page 3	HS		Par. 3.2 - added "(190 psia min)"
		page 5	HS		Par. 4.1.2 +160°F was +1602F
		page 5	HS		Par. 4.1.4.2 27.5 was 31.0
		page 6	HS		Figure 1 512°R was 520°R
		page 4	HS		Added paragraph 3.3.4
#2	Oct. 1972	page 3	HS		Par. 3.2 added 110 psia Min
		page 5	HS		Par. 4.1.1 added liquid hydrogen and Freon; changed hydrogen flow rates.
		page 6	HS		Figure 1 deleted 100 psia

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WYLE LABORATORIES • EL SEGUNDO, CALIF.

SVHSER 6180
TEST PROCEDURE NO. 3474

PAGE NO. 2

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1.0 PURPOSE

THE PURPOSE OF THIS DOCUMENT IS TO PRESENT THE PROCEDURES TO BE EMPLOYED DURING THE PERFORMANCE OF A TEST PROGRAM WHICH WILL BE CONDUCTED TO DETERMINE THE TRANSIENT AND STEADY STATE HEAT TRANSFER CHARACTERISTICS OF ONE HEAT EXCHANGER, PART NUMBER SVSK86021.

2.0 REFERENCES

- 2.1 HAMILTON STANDARD PURCHASE ORDER NUMBER SS043657NL.
- 2.2 HAMILTON STANDARD TEST PLAN AE-2-38.
- 2.3 HAMILTON STANDARD DRAWING NUMBER SVSK86021, DATED 7-20-72.

3.0 TEST CONDITIONS AND EQUIPMENT

3.1 AMBIENT CONDITIONS

UNLESS OTHERWISE SPECIFIED HEREIN, ALL TESTS WILL BE PERFORMED AT AN ATMOSPHERIC PRESSURE OF 29.92 ± 0.5 INCHES OF MERCURY ABSOLUTE, A TEMPERATURE OF $+70 \pm 15^{\circ}\text{F}$, AND A RELATIVE HUMIDITY OF APPROXIMATELY 50 PERCENT.

3.2 TEST MEDIA

All tests will be performed using Supercritical Liquid Hydrogen (190 psia min) (Inner Tube) and liquid Freon 21 (Outer Tube) as the test media. Freon outlet pressure shall be a minimum of 110 psia. 1
2

3.3 INSTRUMENTATION AND EQUIPMENT

- 3.3.1 MEASURING AND TEST EQUIPMENT, UTILIZED IN THE PERFORMANCE OF THIS CONTRACT, WILL BE CALIBRATED BY THE WYLE LABORATORIES STANDARDS LABORATORY, OR A COMMERCIAL FACILITY, UTILIZING REFERENCE STANDARDS (OR INTERIM STANDARDS) WHOSE CALIBRATIONS HAVE BEEN CERTIFIED AS BEING TRACEABLE TO THE NATIONAL BUREAU OF STANDARDS. ALL REFERENCE STANDARDS UTILIZED IN THE ABOVE CALIBRATION SYSTEM, WILL BE SUPPORTED BY CERTIFICATES, REPORTS, OR DATA SHEETS ATTESTING TO THE DATE, ACCURACY, AND CONTROL REQUIRED BY THE SUBJECT CONTRACT.

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3.0 TEST CONDITIONS AND EQUIPMENT (CONTINUED)**3.3** INSTRUMENTATION AND EQUIPMENT (CONTINUED)

3.3.2 WYLE LABORATORIES ATTESTS THAT THE COMMERCIAL SOURCES PROVIDING CALIBRATION SERVICES ON THE ABOVE REFERENCED EQUIPMENT, OTHER THAN THE NATIONAL BUREAU OF STANDARDS, ARE IN FACT CAPABLE OF PERFORMING THE REQUIRED SERVICES TO THE SATISFACTION OF THE WYLE LABORATORIES QUALITY CONTROL DEPARTMENT. CERTIFICATES AND REPORTS OF ALL CALIBRATIONS PERFORMED WILL BE RETAINED IN THE WYLE LABORATORIES QUALITY CONTROL FILES AND WILL BE AVAILABLE FOR INSPECTION, UPON REQUEST, BY CUSTOMER AND/OR NASA REPRESENTATIVES.

3.3.3 THE FOLLOWING TEST EQUIPMENT, OR EQUIVALENT, WILL BE USED DURING THE PERFORMANCE OF THIS PROGRAM.

<u>EQUIPMENT</u>	<u>MFG. & MODEL NO.</u>	<u>RANGE</u>	<u>ACCY.</u>
PRESSURE GAUGE	ASHCROFT, 1279D	200 PSIG	±0.5% FS
PRESSURE GAUGE	ASHCROFT, 1279D	500 PSIG	±0.5% FS
ΔP TRANSDUCERS (2)	VALIDYNE, DP15	AS REQUIRED	N.A.
TEMPERATURE, FLOW & PRESSURE RECORDERS (3)	MOSELEY, 7100B	AS REQUIRED	±0.5%
FLOW METERS (2)	FOXBORO	AS REQUIRED	±0.5%

3.3.4 The temperature probes will have a time constant of less than 0.15 seconds in water.

The turbine flow meters will have a time constant less than 25 milliseconds.

Recorder time constant shall be less than 0.5 seconds.

3.4 SYSTEM INSPECTION AND TEST WITNESS

AUTHORIZED HAMILTON STANDARD AND NASA REPRESENTATIVES WILL HAVE THE RIGHT TO INSPECT THE TEST SYSTEM FOR COMPLIANCE WITH REFERENCE 2.1 AND TO WITNESS ALL TESTS TO BE PERFORMED IN ACCORDANCE WITH THIS PROCEDURE.

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4.0 PROCEDURES AND REQUIREMENTS4.1 HEAT TRANSFER TEST

REFERENCE 2.2

- 4.1.1 The Heat Exchanger will be installed in the Flow System shown in Figure 1 and subjected to liquid hydrogen and liquid Freon flow and temperature conditions in accordance with the steady state steps outlined below.

STEP No.	BTU/HR	FREON			HYDROGEN	
		FLOW RATE (LB/HR)	INLET TEMP. (°F)	OUTLET TEMP. (°F)	FLOW RATE (LB/HR)	INLET TEMP. (°F)
1	5,000	1600	+52	+40	3.0	-420
2	20,000	1600	+88	+40	12.0	-420
3	35,000	1600	+125	+40	21.0	-420
4	50,000	1600	+160	+40	30.5	-420

- 4.1.2 THE TEST DESCRIBED IN PARAGRAPH 4.1.1 WILL BE REPEATED EXCEPT THAT THE FREON INLET TEMPERATURE TRANSITION FROM +52 TO +160°F WILL BE ACCOMPLISHED AT A RATE OF $20 \pm 5^\circ\text{F}$ PER MINUTE.

- 4.1.3 THE TEST DESCRIBED IN PARAGRAPH 4.1.1 WILL BE REPEATED EXCEPT THAT THE FREON INLET TEMPERATURE TRANSITION FROM +52 TO +160°F WILL BE ACCOMPLISHED IN ONE STEP AT THE MAXIMUM SYSTEM CAPABILITY.

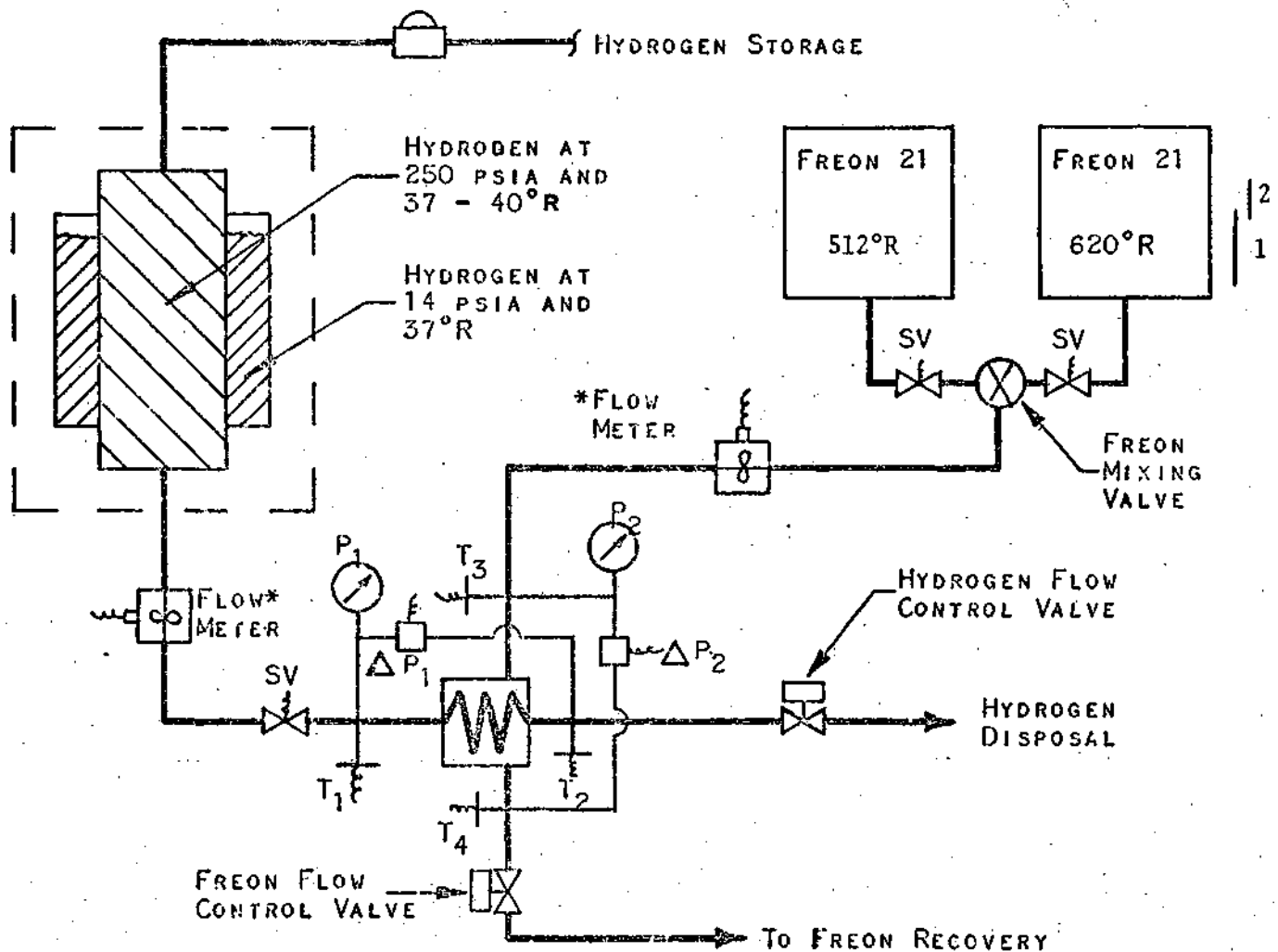
- 4.1.4 DURING THE STEADY STATE RUNS DESCRIBED IN PARAGRAPH 4.1.1, FREON OUTLET TEMPERATURE TRANSIENTS WILL BE DETERMINED AS FOLLOWS:

- 4.1.4.1 AFTER STEP 1 STEADY STATE CONDITIONS HAVE BEEN ESTABLISHED, HYDROGEN FLOW WILL BE STOPPED FOR APPROXIMATELY 5 MINUTES. THE HYDROGEN FLOW RATE OF 3.0 POUNDS PER HOUR WILL THEN BE INSTANTLY ESTABLISHED AND MAINTAINED UNTIL THE FREON OUTLET TEMPERATURE REACHES +40°F.

- 4.1.4.2 AFTER STEP 4 STEADY STATE CONDITIONS HAVE BEEN ESTABLISHED, THE HYDROGEN FLOW WILL BE SHUTOFF FOR APPROXIMATELY 5 MINUTES. THE HYDROGEN FLOW RATE OF 30.5 POUNDS PER HOUR WILL THEN BE INSTANTLY ESTABLISHED AND MAINTAINED UNTIL THE FREON OUTLET TEMPERATURE REACHES +40°F. THE HYDROGEN FLOW WILL THEN BE STOPPED AND THE FREON OUTLET TEMPERATURE WILL BE RECORDED UNTIL STABILIZATION OCCURS.

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FIGURE 1
FREON/HYDROGEN FLOW SYSTEM

**LEGEND**

- | | |
|--|---|
| P_1 - HYDROGEN INLET PRESSURE GAUGE (0-500 PSIG) | $*T_1$ - HYDROGEN INLET TEMPERATURE PROBE (PLATINUM) |
| $*\Delta P_1$ - HYDROGEN ΔP TRANSDUCER | $*T_2$ - HYDROGEN OUTLET TEMPERATURE PROBE (PLATINUM) |
| P_2 - FREON INLET PRESSURE GAUGE (0-200 PSIG) | $*T_3$ - FREON INLET TEMPERATURE PROBE (PLATINUM) |
| $*\Delta P_2$ - FREON ΔP TRANSDUCER | $*T_4$ - FREON OUTLET TEMPERATURE PROBE (PLATINUM) |
| SV - SOLENOID VALVE | |

*THESE PARAMETERS WILL BE CONTINUOUSLY RECORDED DURING TEST.

APPENDIX B

INSTRUMENTATION LIST

SPECIMEN HEAT EXCHANGER
 CUSTOMER HAMILTON STANDARD
 PART NO. SVSK 86021
 S/N NONE

JOB NO. 53163
 DATE 1-10-73
 TEST BY R. J. Pappert
 WITNESS J. Butler

WYLE LABORATORIES

TEST: HEAT TRANSFER

EQUIPMENT	MANUFACTURER	MODEL NO.	RANGE	WYLE NO.	CALIBRATION		ACCY.
					LAST	DUE	
OSCILLATOR	HEWLETT PACKARD	200C DR	5HZ TO 600K HZ	30819	7-13-72	1-14-73	+2% FREQ.
MILLIVOLT POTENTIOMETER	HONEYWELL	2745	0.01 TO 800 M.V.	2259	11-7-72	3-11-73	+0.3%
DECADE RESISTOR	ELECTRO SCIENTIFIC	DB52	0.01 TO 1,111.1 Ω	31499	11-27-72	5-27-73	+0.05%
THERMOCOUPLE REF JUNCTION	PAGE ENGINEERING	825C13A-12TP	+150°F REFERENCE	31456	11-4-72	5-6-73	±0.15°F
MOSELEY RECORDER	HEWLETT PACKARD	7100B	2" SEC. TO 1" HOUR	31160	10-27-72	2-25-73	N/A
MOSELEY AMPLIFIER	HEWLETT PACKARD	17501A	1 M.V. TO 100 V.	31237	9-16-72	1-21-73	±0.2% F.S.
MOSELEY AMPLIFIER	HEWLETT PACKARD	17501A	1 M.V. TO 100 V.	31486	10-24-72	2-25-73	±0.2% F.S.
MOSELEY RECORDER	HEWLETT PACKARD	7100B	2" SEC. TO 1" HOUR	31236	10-27-72	2-25-73	N/A
MOSELEY AMPLIFIER	HEWLETT PACKARD	17501A	1 M.V. TO 100 V.	31217	10-26-72	2-25-73	±0.2% F.S.
MOSELEY AMPLIFIER	HEWLETT PACKARD	17501A	1 M.V. TO 100 V.	31282	10-25-72	2-25-73	±0.2% F.S.
MOSELEY RECORDER	HEWLETT PACKARD	7100B	2" SEC. TO 1" HOUR	31132	10-25-72	2-25-73	N/A
MOSELEY AMPLIFIER	HEWLETT PACKARD	17501A	1 M.V. TO 100 V.	31214	10-27-72	2-25-73	±0.2% F.S.
MOSELEY AMPLIFIER	HEWLETT PACKARD	17501A	1 M.V. TO 100 V.	31215	10-27-72	2-25-73	±0.2% F.S.
MOSELEY RECORDER	HEWLETT PACKARD	7100B	2" SEC. TO 1" HOUR	31485	10-20-72	2-25-73	N/A
MOSELEY AMPLIFIER	HEWLETT PACKARD	17501A	1 M.V. TO 100 V.	31487	9-16-72	1-21-73	±0.2% F.S.
MOSELEY AMPLIFIER	HEWLETT PACKARD	17501A	1 M.V. TO 100 V.	31216	10-27-72	2-25-73	±0.2% F.S.
TEMPERATURE BRIDGE	ROSEMOUNT ENG.	421B	10 CHANNEL	31514	SYSTEM	CALIBRATION	N/A

W 614 C

Q.C. Approval Am

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SVHSR 6180

Page No. 13

SPECIMEN HEAT EXCHANGER
 CUSTOMER HAMILTON STANDARD
 PART NO. SVSK 86021
 S/N NONE
 TEST: HEAT TRANSFER

JOB NO. 53163
 DATE 1-10-73
 TEST BY [Signature]
 WITNESS J. Butler

WYLE LABORATORIES

EQUIPMENT	MANUFACTURER	MODEL NO.	RANGE	WYLE NO.	CALIBRATION		ACCY.
					LAST	DUE	
ELECTRONIC COUNTER	HEWLETT PACKARD	5233L	1 μSEC. TO 10 SEC.	31497	11-17-72	3-18-73	±1 COUNT
FREQUENCY METER	HEWLETT PACKARD	500BR	10 HZ TO 100 KC	31524	11-19-72	3-11-73	±2%
AMPLIFIER	ENDEVCO	2704	5 HZ TO 10 KHZ	30371	8-7-72	2-11-73	±2% F.S.
CARRIER DEMODULATOR	VALIDYNE	CD15	0-10 VDC	31609	SYSTEMS	CALIBRATION	N/A
PRESSURE TRANSDUCER	CEC	170522-0100	± 50 PSID	31526 S/A	SYSTEMS	CALIBRATION	N/A
PRESSURE TRANSDUCER	VALIDYNE	DP15TL	± 1 PSID	6342	SYSTEMS	CALIBRATION	N/A
DIGITAL VOLTMETER	DYNEL	2401C	0-1000 V.D.C.	31284	1-5-73	5-6-73	±0.1%
TEMPERATURE POTENTIOMETER	LEEDS NORTHROP	8686	-340°F TO +230°F	2249	9-21-72	1-21-73	±0.2% F.S.
TEMPERATURE PROBE	ROSEMOUNT	150MA12	0°C TO -268.944°C	31518 S/N	1-4-73	1-4-74	±1°C
FLOWMETER	WAUGH	FL-850	50 HZ TO 500 HZ	13437	12-29-72	12-29-73	±1%
INCLINE MANOMETER	ELLISON	144	0-4 IN. H ₂ O	2500	11-27-72	5-27-73	±0.1" H ₂ O
MANOMETER	MERIAM	30R335	0-60 IN Hg	30068	9-26-72	3-25-73	±0.1" Hg
MANOMETER	MERIAM	30EB25	0-70 IN H ₂ O	3713	12-4-72	6-3-73	±0.1" H ₂ O
MANOMETER	MERIAM	30EB25	0-70 IN H ₂ O	30064	8-11-72	2-11-73	±0.1" H ₂ O
GAUGE - PRESSURE	ASHCROFT	12798	0-1000 PSI	31186	12-16-72	2-9-73	±0.5%
GAUGE - PRESSURE	MARSH	12795	0-500 PSI	31631	9-8-72	2-3-73	±0.5%
GAUGE - PRESSURE	LONERGAN	4A-301	0-200 PSI	4259	11-19-72	2-3-73	±0.5%

W 614 C

O.C. Approval [Signature]

SHEET OF

JOB NO. 59163
DATE 1-10-73
TEST BY Whisper
WITNESS C. Butler

TEST: HEAT TRANSFER

* Used only on 1-9-72	73 AH
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Q.C. Approval *Ans*

SHEET OF

APPENDIX C

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